COMPARISON OF STRATIFIED AND HOMOGENEOUS COMBUSTION IN A DIRECT-INJECTED TWO-STROKE ENGINE FOR SNOWMOBILE APPLICATIONS

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ABSTRACT

The University of Idaho has been developing gasoline direct injection two-stroke engine technology for use in snowmobile applications as part of the Clean Snowmobile Challenge organized by the Society of Automotive Engineers. Applying GDI to a two-stroke engine significantly reduces emissions of unburned hydrocarbons and improves fuel economy by reducing or eliminating the short circuiting of fuel that occurs in conventional carbureted two-stroke engines. The University of Idaho's GDI design allows for two different modes of combustion, stratified and homogeneous. Stratified combustion is used during idle and light to moderate loads at low engine speeds while homogeneous combustion used at moderate to high loads and medium to high engine speeds. During calibration work there has been debate on where the transition from stratified to homogeneous combustion should occur and which mode of combustion provides better fuel economy during cruise point operation of the engine.

This thesis presents the process and results of determining which mode of combustion provides better fuel economy during cruise point operation, and where the transition from stratified to homogeneous combustion should occur. Furthermore, results are presented regarding the effects of increased squish velocity on the brake specific fuel consumption and torque output during stratified operation. Additionally, this thesis is intended to provide a basic understanding and background of two-stroke engine theory, GDI systems, and stratified and homogeneous combustion to new students on the Clean Snowmobile team.

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DEFINITION OF TERMS

GDI:	Gasoline Direct Injection
UHC:	Unburned Hydrocarbons
SAE:	Society of Automotive Engineers
CSC:	Clean Snowmobile Challenge
BSFC:	Brake Specific Fuel Consumption
CO:	Carbon Monoxide
NO_X:	Nitrogen Oxides
EPA:	Environmental Protection Agency
NPS:	National Park Service
SwRI:	Southwest Research Institute
Ppm:	Parts per million
g/hr:	grams per hour
g/kW-hr:	grams per kilowatt-hour
CVT:	Continuously Variable Transmission
BDC:	Bottom Dead Center
TDC:	Top Dead Center
BTDC:	Before Top Dead Center
ATDC:	After Top Dead Center
WOT:	Wide Open Throttle
EGR:	Exhaust Gas Residuals
CDI:	Capacitive Discharge Ignition
AFR:	Air Fuel Ratio
SMD:	Sauter Mean Diameter
MBT:	Maximum Brake Torque
COV of IMEP:	Coefficients of Variance of Indicated Mean Effective Pressure

1.0 INTRODUCTION

Over the past decade, the emissions regulations for transportation vehicles, recreational vehicles, and stationary power generating equipment have become more stringent. With these lower emissions requirements and the improvements in technology, there has been an increase in the amount of research being conducted on fuel delivery systems, combustion systems, and exhaust after-treatment techniques in order to meet new regulations.

One area of research that has seen extensive activity over the past few years is the development and application of Gasoline Direct Injection (GDI) systems to a variety of engines ranging from automotive applications to the recreational industry. A typical GDI system injects high pressure fuel directly into the cylinder through the cylinder head. The recent development of this technology is very appealing to the designers and manufacturers of two-stroke engines, where large reductions in unburned hydrocarbon emissions (UHC) and great improvements in fuel economy can be had by implementing GDI.

The University of Idaho has been researching the application of GDI to two-stroke engines for snowmobile applications since 2003. This research is being done as part of an ongoing student design project for the Society of Automotive Engineers (SAE) Clean Snowmobile Challenge (CSC). The research presented in this thesis stems from the SAE CSC project and addresses some questions about engine cruise point calibration strategies.

The GDI system implemented at the University of Idaho allows for two different modes of combustion, stratified and homogeneous. The stratified mode of combustion allows the engine to operate in a similar manner as a diesel engine where the fuel vaporizes as combustion occurs and the fuel-air mixture in the cylinder is not thoroughly mixed. This allows for reduced fuel consumption at idle and low load conditions, along with the ability to run the engine un-throttled. The homogeneous mode of combustion allows the engine to operate like a typical gasoline engine, with a thoroughly mixed fuel-air mixture in the cylinder. Homogeneous combustion is necessary for moderate to high loads and peak power. During cruising conditions, the engine is operated at part throttle, not quite moderate load, but not light load either. Under these conditions both stratified and homogeneous combustion provide the required power for the snowmobile to operate. One research question is: which mode of combustion is more efficient during cruising conditions? A second research question is: where in the calibration should the transition from stratified to homogeneous occur?

1.1 THESIS GOALS

These two questions are the primary focus of this thesis. The first purpose of this research is to compare both stratified and homogeneous combustion throughout the cruise range of the engine and determine which mode of combustion provides better fuel economy. The second purpose is to identify at which throttle settings and engine speeds the transition from stratified to homogeneous combustion should occur. The third purpose is an investigation into the effects of squish velocity on stratified combustion in an attempt to improve both the maximum achievable power output and fuel economy during stratified operation. For all of these tests, the primary evaluation criteria were brake specific fuel consumption (BSFC), power output, and run quality, or misfire rate of the engine.

1.2 CLEAN SNOWMOBILE CHALLENGE

The Clean Snowmobile Challenge was started in 2000 in an attempt to promote the development of clean and quiet snowmobiles [1]. The goal of the competition is to reduce unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions without increasing nitrogen oxides (NO_X) emissions and reduce noise output while maintaining or improving factory power output and handling characteristics. The CSC event targets the problem of operating snowmobiles in environmentally sensitive areas such as Yellowstone National Park. Currently there are two emissions standards for snowmobiles: the 2012 Environmental Protection Agency (EPA) standards and the National Park Service (NPS) standards. The 2012 EPA standards must be met as a corporate average by all snowmobile manufacturers by 2012. The NPS standards are stricter than the EPA standards and are required for a snowmobile to operate in Yellowstone, Grand Teton, or Rockefeller Memorial Parks [2]. To determine whether a snowmobile meets the EPA or NPS standards, a weighted five mode test developed by the Southwest Research Institute (SwRI) is performed on the engine, which represents the duty cycle for snowmobile engines while in operation [1]. During the test, emissions measurements are taken at five different engine speeds and five different power outputs, with each mode point having a weighting applied as detailed in Table 1 [1].

Mode Point	***1***	***2***	***3***	***4***	***5***
Speed (% of Rated)	100	85	75	65	Idle
Torque (% of Rated)	100	51	33	19	0
Weighting Factor (%)	12	27	25	31	5

Table 1: *The five modes used for snowmobile testing for the EPA and NPS.*

The emissions measured during the five mode test are carbon monoxide (CO), unburned hydrocarbons (UHC) and oxides of nitrogen (NO_X). The emissions are measured in a concentration of either parts per million (ppm) or percent concentration. Then they are converted to a mass flow rate with units of grams per hour (g/hr). To get brake specific emissions used in the standards, the mass flow rate of emissions for each mode point is divided by the power produced at that mode point, yielding brake specific emissions with units of grams per kilo-Watt-hour (g/kW-hr). The brake specific emissions measured at each mode point are then multiplied by their respective weighting factor from the table above, and the resulting products are added together to achieve a final weighted brake specific emissions value for UHC, CO, and NO_X. These numbers are then used in equation 1.1.1 to determine the sled emission number *E*.

$$E = \left[1 - \frac{(HC + NO_x) - 15}{150}\right] * 100 + \left[1 - \frac{CO}{400}\right] * 100 \qquad (1.1.1)$$

The 2012 EPA standard requires a minimum score of 100 while the NPS standard requires a stricter score of 170. In addition to the minimum score for *E*, both standards require that UHC+NO_X and CO emissions not exceed a maximum value. For the 2012 EPA standard, UHC+NO_X must not exceed 90 g/kW-hr and CO emissions must not exceed 275 g/kW-hr. The NPS standard requires that UHC+NO_X not exceed 15 g/kW-hr and CO emissions must not exceed 120 g/kW-hr. These requirements are outlined in Table 2 [1].

Standard	**CO (g/kW-hr)**	**HC+NO_X (g/kW-hr)**
EPA	275	90
NPS	120	15

Table 2: *Maximum emissions limits for EPA and NPS standards.*

The University of Idaho has participated in the CSC event each year since 2001. The first three years Idaho entered a 2001 Arctic Cat SnoPro chassis with a 1991 BMW K75RT motorcycle engine which won the CSC in 2002 and 2003 and scored 199 on the E scale. In 2004 Idaho switched to a two-stroke direct injection design having proved that a four-stroke is capable of meeting the NPS standards.

The snowmobile market is targeted for mostly recreational use with consumers preferring low weight, good handling and high power. The performance of a snowmobile is directly related to the weight of the vehicle and the power output of the engine. Snowmobiles need to handle well, accelerate well and ultimately provide the rider with an enjoyable experience. The two-stroke engine has a high power-to-weight ratio, is small in size when compared to four-stroke engines of the same power output, and has a power curve that is well suited for the continuously variable transmission (CVT) type drivetrain found in snowmobiles. However, the two-stroke engine has historically suffered from poor fuel economy and high levels of UHC emissions. As noted in the previous section, stricter emissions regulations are affecting the recreational vehicle market, forcing manufacturers to develop cleaner engine technologies for these markets. This usually means using a four-stroke engine in place of a two-stroke for the benefit of decreased emissions and improved fuel economy. This solution path has been proven time and again through the SAE CSC. From 2000-2006 all CSC winners were powered by four-stroke engines with catalytic converter after-treatment. As mentioned in the previous section, Idaho has won two of these years by retrofitting a four-stroke engine into a snowmobile chassis. However, these modifications sacrificed power and increased weight over a similar two-stroke design.

With the recent improvements fuel delivery systems, specifically direct injection, the potential for improved emissions and fuel economy in two-stroke engines is greatly enhanced. Evinrude outboard engines have reliably and repeatedly proven that GDI two-stroke engine technology is capable of emissions levels equivalent to that of four-stroke engines [3]. With these data, the University of Idaho decided to take on the challenge of implementing direct injection to a two-stroke snowmobile engine for the CSC in an attempt to meet the 2012 EPA standards and to eventually meet the NPS standard without sacrificing the high power output and low weight that are preferred by the majority of snowmobile riders [4,5,6].

2.0 THE TWO-STROKE ENGINE

The two-stroke engine is mechanically simpler than a four-stroke engine since the only moving parts are the piston, connecting rod, and crankshaft. This is because airflow in a two-stroke engine is controlled by ports in the cylinder wall and the location of the piston instead of a valvetrain and passages in the cylinder head of the four-stroke engine. The two-stroke engine is also able to intake a fuel air charge, compress it, combust it, and expel the combustion gases with only two-strokes of the piston - or one crankshaft revolution. A four-stroke engine requires four-strokes of the piston - or two crankshaft revolutions. Since the two-stroke engine produces a power pulse on every downward stroke of the piston as opposed to every other downward stroke of the piston in a four-stroke engine, the two-stroke engine can produce nearly twice the power of an identically sized four-stroke engine. While the two-stroke engine is mechanically simple, the airflow through it is not, as will be shown in the following description of a reed-valved, crankcase charged two-stroke engine.

2.1 THE TWO-STROKE ENGINE CYCLE

The engine cycle begins with the piston at bottom dead center (BDC) as shown in Figure 1. With the piston in this location, the scavenging ports and exhaust ports are fully open and the scavenging process is half way complete. The crankcase volume is also at a minimum.

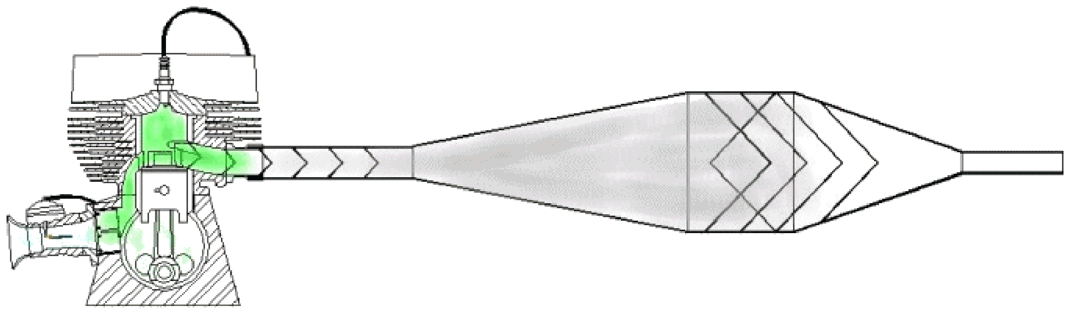

Figure 1: *Piston at BDC with both the scavenging and exhaust ports open.*

As the piston moves away from BDC the scavenging ports begin to close and the flow of fresh air from the crankcase decreases because the volume of the crankcase is increasing, diminishing the pressure differential driving the flow. In these figures air is shown in blue color, the unburned air/fuel mixture is shown in green, and the burned combustion products are shown in grey. Once the intake ports close, the incoming charge through the scavenging ports stops. It is at or near this point that a reflected pressure wave, created by the initiation of blow-down of the previous cycle, enters the exhaust port and the cylinder (Figure 2). This wave is reflected by the tuned exhaust pipe and increases the air density in the cylinder before the piston covers the exhaust port.

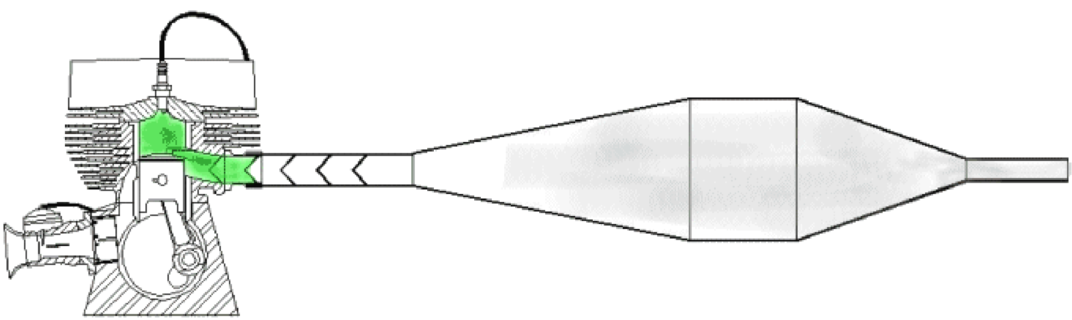

Figure 2: *Scavenging ports closed, plugging pulse arriving at cylinder.*

When the piston covers the exhaust port the compression part of the upward stroke starts. The air and fuel in the cylinder at this point is referred to as the trapped charge. While the piston moves up from BDC towards top dead center (TDC), the crankcase volume is expanding. This expansion creates a pressure drop across the reed valves located in the wall of the crankcase. Once the pressure differential is great enough, the reed valves open and crankcase filling begins (Figure 3).

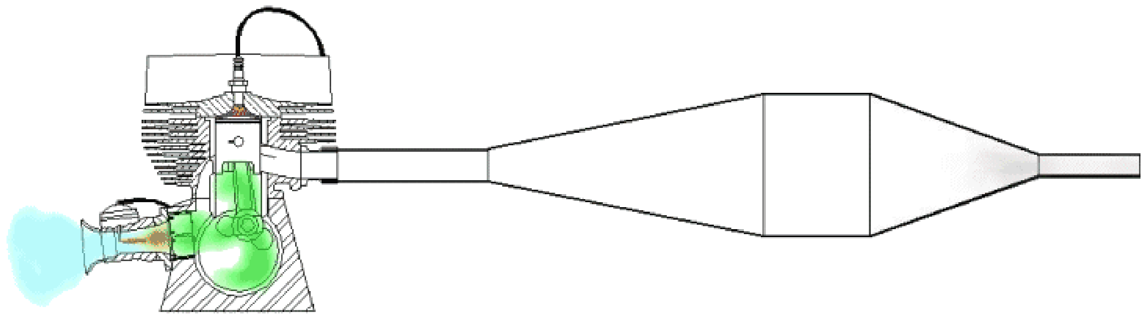

Figure 3: *Fuel Air mixture compression and crankcase expansion.*

About 12-25 degrees before top dead center (BTDC) the spark plug ignites the fuel and air mixture causing the cylinder pressure to rise until it peaks at about 10 degrees after top dead center (ATDC). When the piston is at TDC, the crankcase volume is at a maximum and air flow through the reed valves has slowed down. As the piston starts to move down, the volume of the crankcase decreases, increasing the crankcase pressure and causing the reed valves to close. Once the reed valves close, compression of the crankcase charge begins (Figure 4).

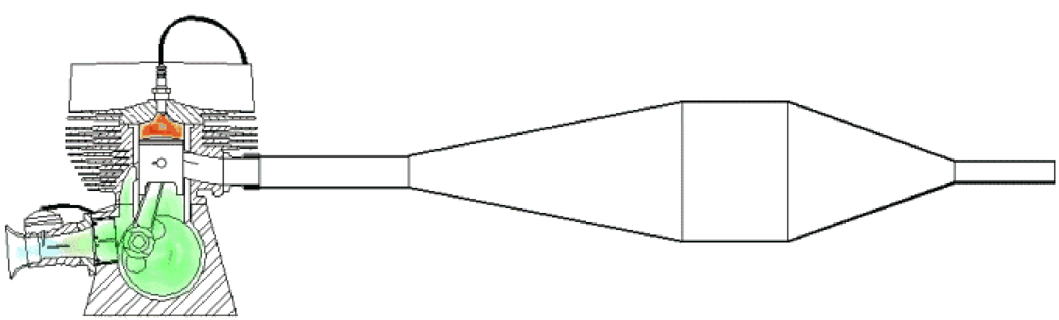

Figure 4: *Combustion of the fuel/air mixture and start of crankcase compression.*

When the top of the piston has reached the top of the exhaust ports (Figure 5) the blow-down process begins. Blow-down is the start of the exhaust purging process where the high pressure combustion gases are released into the exhaust port. The opening of the exhaust port causes a pressure wave to emanate into the tuned pipe portion of the exhaust system. This wave is later reflected back as a plugging pulse to the cylinder on the next cycle (Figure 2).

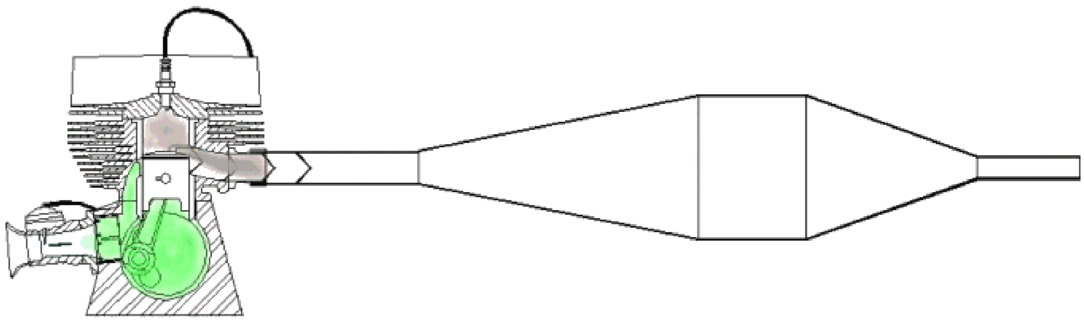

Figure 5: *Start of the blow-down process and purging of combustion gases.*

As the piston continues towards BDC, the scavenging ports are uncovered and the compressed charge in the crankcase flows through the scavenging ports and into the cylinder (Figure 6). This marks the start of the scavenging process where the fresh incoming charge fills the cylinder and displaces the combustion gases out of the exhaust port.

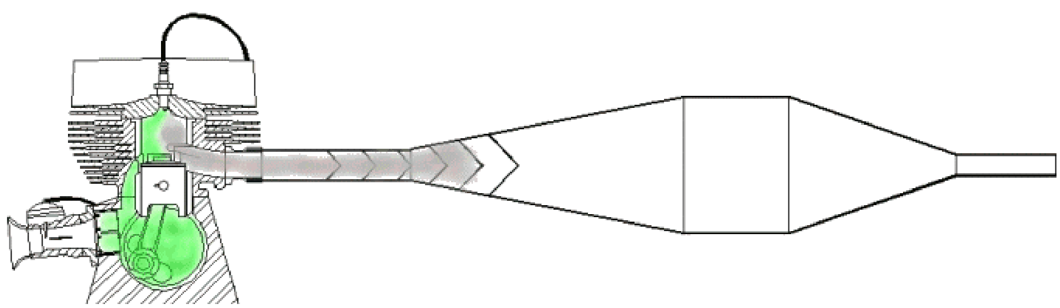

Figure 6: *Transfer port opening and start of the scavenging process.*

It should be noted that some short-circuiting of the charge occurs into the first part of the exhaust port and pipe (Figure 7) and that not all of the combustion gases are scavenged from the cylinder. However, some of this short circuited charge is pushed back into the cylinder by the reflected pressure wave created by the tuned pipe. Finally, as the piston reaches BDC, the scavenging flows slow down and the cycle starts over.

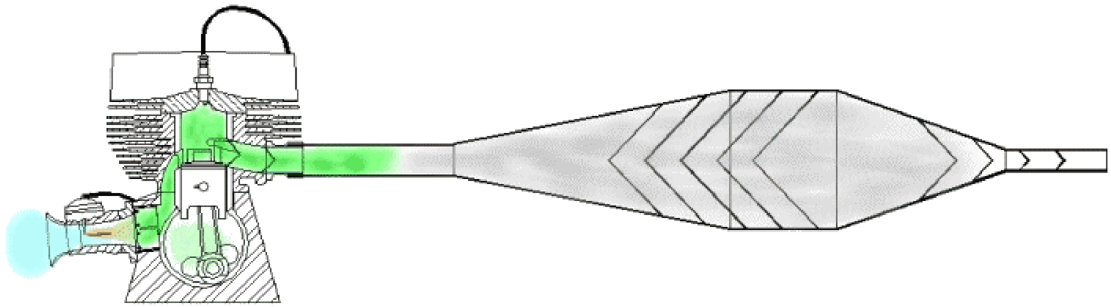

Figure 7: *Short circuiting of the scavenging charge into the exhaust port and pipe.*

2.2 ENGINE SELECTION

The engine selected by Idaho for modification, and used for data collection, is a 2002 Polaris Liberty 600 with specifications shown in table 3.

Bore	77 mm
Stroke	64 mm
Rod	124 mm
Displacement	600 cc
Exhaust Port Timing	102 ATDC
Exhaust Port Duration	156°
Intake Port Timing	120 ATDC
Intake Port Duration	120°
Scavenging Period	120°
Trapped Compression	5.92
Geometric Compression	8.38

Table 3: *Engine Specifications*

The engine is a twin cylinder two-stroke engine displacing 599 cc and is carbureted in stock configuration. The engine is best classified as a reed valved, crankcase charged, and Schnurle loop scavenged two-stroke with variable exhaust and a tuned pipe. A cross section of a similar engine is shown in Figure 8 [7].

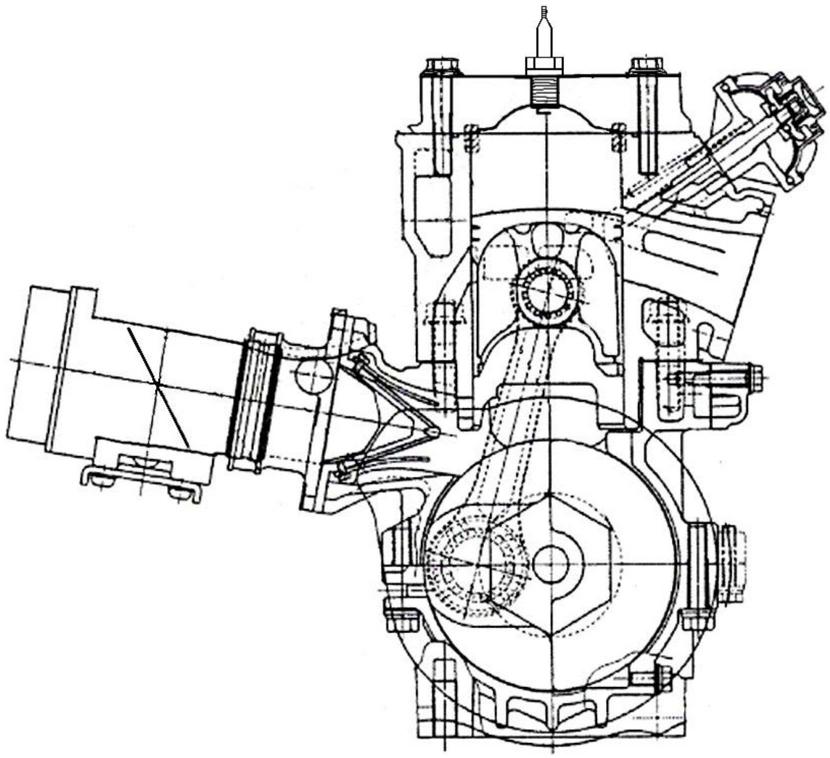

Figure 8: *Cross Section of an engine similar to the one used in testing [7].*

A Schnurle loop engine utilizes one exhaust port and two scavenging ports that are located around the cylinder symmetric to the exhaust port and on the same level as the exhaust port as shown in Figure 9 [8, 9]. During the scavenging process, the placement of these ports causes the incoming fresh charge to be directed upward into the cylinder and not directly at the exhaust port, eliminating the need for a deflector style piston and offering better scavenging performance at throttled conditions than other port configurations. [8]. However, at high loads and high speeds, the scavenging performance of a Schnurle loop engine is not as good as other port configurations.

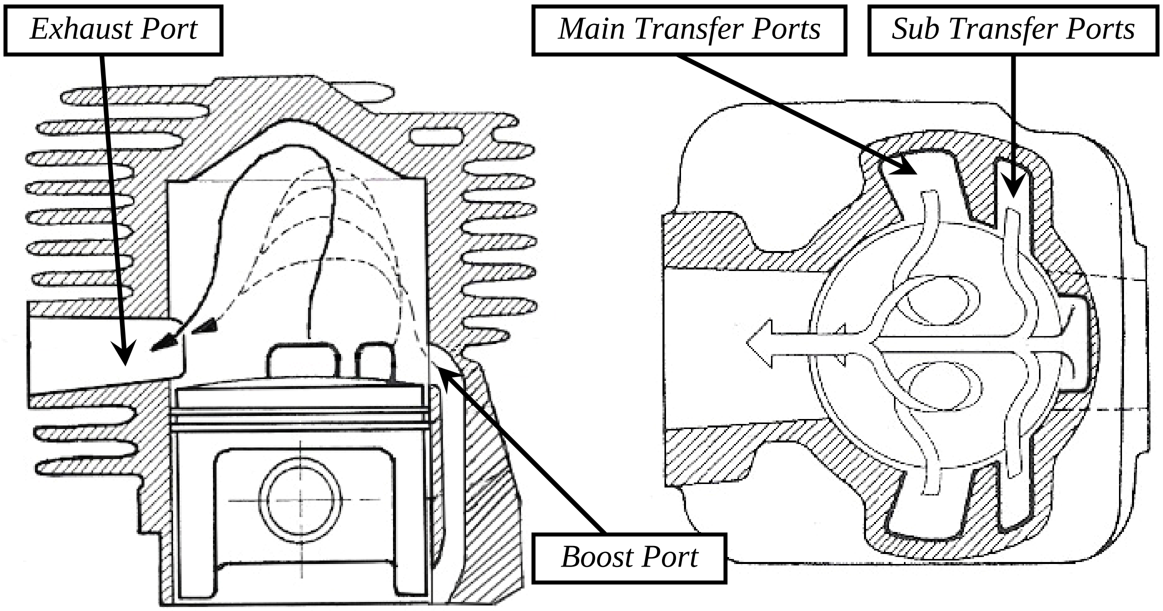

Figure 9: *Schematic of in-cylinder flows for a Schnurle loop-scavenged engine [8, 9].*

The reduced scavenging performance at high load and high engine speeds can be improved through the use of variable exhaust port timing. This test engine uses exhaust valves that are controlled by a pressure difference between the in-cylinder combustion pressure and atmospheric pressure. When the engine is under low to medium engine speeds, the cylinder pressure is not great enough to open the exhaust valves. When they are in this closed position, they effectively reduce the height of the exhaust port, offering a higher trapped compression ratio, longer power stroke, less time for short circuiting of fuel and air, and as a result, improved mid range torque, power and emissions. Once the engine reaches wide open throttle (WOT) and a speed greater than ~6500 rpm, the in-cylinder pressures become high enough to open the exhaust valves, raising the exhaust port timing. This reduces the trapped compression ratio and shortens the expansion stroke, but the tuning of the pipe results in a higher trapped mass in the cylinder at these operating conditions, making the dynamic compression of the engine higher than at low load conditions. The raised port timing also increases the scavenging performance of the engine by starting the blow-down process earlier in the cycle, allowing a longer period of time for the scavenging event to take place. For the test engine, exhaust port timing is 102 degrees after top dead center (ATDC) with the power valves closed and is increased to 85 degrees ATDC when the valves are open.

Another improvement in high load, high speed performance can be achieved through the use of a tuned pipe. As mentioned earlier, a tuned pipe reflects pressure pulses created at the start of the blow-down process back to the exhaust port, pushing some or all of the short circuited fuel and air charge (along with exhaust gas residuals) back into the cylinder. This increases the trapped mass in the cylinder at exhaust port closure and results in a higher dynamic compression ratio, increasing engine efficiency and power output. However, the benefits of a tuned exhaust pipe are only seen over a small range of engine speeds. This occurs because the pipe must be tuned to reflect pressure waves so they are timed to reach the cylinder after the scavenging ports are closed but before piston closes the exhaust port. Outside of the designed operating engine speeds, the exhaust pipe will negatively affect engine performance by providing untimely positive pressure waves containing high amounts of exhaust gas residuals (EGR), instead of short circuited air and fuel, into the cylinder. This wave of EGR dilutes the fresh charge, causing combustion instability that varies from cycle to cycle and degrades power output and engine run quality. However, these negative effects are also only seen over a narrow range of operating speeds.

In a carbureted two-stroke engine, the fuel is mixed with the air as it enters the crankcase, thus the scavenging process uses the incoming fuel-air charge to purge the exhaust gas residuals out of the cylinder. During this process some of the fuel-air charge is short-circuited through the exhaust port. This is pronounced at off-design operating conditions and at WOT when the engine is operated in a fuel-rich condition for maximum power. The short circuiting of fuel is what accounts for high UHC emissions and poor fuel economy. Short circuiting can account for up to 50 percent of the supplied fuel [9]. However, the CVT used in snowmobiles keeps the two-stroke engine close to design speeds and loads, reducing the amount of short circuited fuel to only 10-30 percent [9].

Short circuiting can be drastically reduced if not eliminated in most operating conditions of a two-stroke by using GDI. A GDI system injects fuel directly into the cylinder through the cylinder head allowing the scavenging process to take place without fuel in the incoming charge. Precise computer control of the timing of fuel injection event, quantity of fuel injected, and the timing of the ignition event allow a two-stroke engine to operate with minimal short circuiting at high speeds and loads and eliminates short circuiting at low speeds and loads. However, applying GDI to a two-stroke engine will not eliminate the poor run quality at off-design operating conditions due to EGR from poor scavenging performance [10].

3.0 GASOLINE DIRECT INJECTION

3.1 BENEFITS

The two primary benefits of implementing a GDI system on a two-stroke engine are to a) control the timing of the fuel delivery event such that short-circuiting of the fuel is minimized or eliminated, and b) to allow the engine to run in a stratified mode of combustion for reduced fuel consumption and emissions at low engine speeds and loads. However, these benefits also come with an increase in the complexity of the engine control system.

A GDI system is much more complex than a carbureted engine or even a port fuel injected engine as there is an increase in not only the electronics required for the system, but also in the number and range of calibration variables required to achieve optimum engine performance. In a typical GDI system, the engine computer will monitor throttle position, manifold pressure, engine temperature, air temperature, and engine speed while varying the injection timing, ignition timing, and fuel quantity, all of which were determined through a detailed mapping process. Another difficulty lies with the design of a combustion system that provides stable operation over a wide range of operating conditions while allowing for two different modes of combustion. One particular difficulty is the control of stratified combustion over a wide operating range and a smooth transition from stratified to homogeneous combustion across load changes [11]. This is one of the foci for this research and the results are discussed in subsequent sections.

One design parameter that is key to the robustness of a GDI system is the relative location of the spark plug to the injector. The spark plug must be in a zone of an ignitable and combustible mixture for all operating conditions and must provide a robust combustion process capable of delivering stable engine operation. If charge stratification is desired, then careful attention must be given to the method used to achieve stratification so that homogeneous operation does not deteriorate. Another important consideration in any GDI system is the type of injector used and how its particular fuel spray will affect performance. Achieving a robust system capable of stable stratified operation and powerful homogeneous operation is not easy. In-cylinder air motion created by swirl and tumble, along with fuel spray characteristics such as droplet size, spray cone angle, and calibration parameters such as injection and ignition timing all affect the ability of a system to operate over a wide range of engine operating conditions. A GDI system that is operable over a wide range of conditions is usually a result of compromises on several designs, that when put together offer acceptable performance for all engine conditions [11].

3.2 GDI CLASSIFICATIONS

While the concept of direct injection appears simple, there are numerous ways of implementing a GDI system to an engine, each having special characteristics that need careful attention during the design process. The following is a brief description of three major classifications used to describe a GDI system. They are as follows: the location of the spark plug relative to the injector, the method used for charge stratification, and the injector type. Table 4 is a summary of all relevant GDI classifications for two-stroke engines [9, 11]. A more detailed description that focuses on the specific system used in this research will be presented later in section 3.3.

Category	Classification	Description
Distance between spark plug gap and injector tip	Narrow-Spacing	The spark plug gap is close to the injector tip in order to ignite the spray periphery directly. Also called spray-guided.
	Wide-Spacing	A relatively large distance between the spark plug and injector tip. Stratification is a result of air-flow or spray/wall impingement. Also called air-guided or wall-guided
Approach to creating a stratified charge	Spray-Guided	Stratification results from spray penetration and mixing. Spark plug gap is close to the injector tip and is a narrow spacing concept.
	Wall-Guided	Fuel spray is directed towards a shaped cavity in the piston crown with stratification resulting form this interaction.
	Air-Guided	Stratification is achieved through the use of air-assisted injectors and/or the interaction between the fuel spray and in-cylinder air flows and is usually a wide spacing concept.
Charge Motion	Tumble-Based	Tumble is used to create or assist in charge stratification
	Swirl-Based	Swirl is used to create or assist in charge stratification
	Squish-Based	Squish is used to localize stratified fuel distribution near the spark plug (not typically used in four-stroke applications)
Injector Location	Centrally-Mounted	Injector is located near the center of the combustion chamber
	Side-Mounted	Injector is located in the periphery with the spark plug near the center
Injector Type	Single-Fluid	Single-Liquid fuel is injected at high pressure
	Air-Assisted	A mixture of air and fuel is injected using moderate to low pressures
Fuel Distribution	Homogeneous	Maximum air utilization and a homogeneous or nearly homogeneous air/fuel mixture exists throughout the cylinder
	Stratified	Highly stratified charge is used with varying degrees of AFR throughout the cylinder, locally rich near the spark plug gap.
Injection Timing	Early Injection	Fuel is injected late in the expansion stroke or early in the compression stroke, some stratification may still result.
	Late Injection	Fuel is injected just before or after the exhaust ports are closed to form a highly stratified air/fuel mixture.
Air/Fuel Ratio	Rich	The trapped air/fuel ratio is rich of stoichiometric, regardless of charge stratification.
	Stoichiometric	The trapped air/fuel ratio is stoichiometric, regardless of charge stratification.
	Lean	The trapped air/fuel ratio is lean of stoichiometric. This can be a result of a homogeneous-lean mixture or a stratified (locally rich) mixture.

Table 4: *Summary of relevant GDI classifications for two-stroke engines [9, 11]*

3.2.1 Spark Plug Location

There are two classifications for the relative placement of the spark plug to the injector. In either case, the location of the spark plug is usually kept in the center of the cylinder at the axis of the cylinder to provide symmetric and maximized flame propagation, and to maximize specific power while reducing heat losses and auto-ignition tendencies [11].

The first classification for spark plug and injector location is narrow-spacing. This type of system places the spark plug close to the injector tip and allows the spark plug to directly ignite the fuel spray at its periphery. With a centrally mounted spark plug, the placement of the injector is very close to the center of the combustion chamber. With the narrow-spacing system, charge stratification is achieved by directing the fuel spray towards the plug. This method of stratification is classified as spray-guided and is capable of relatively high stratification of the fuel around the spark plug. Due to the short distance between the injector and the spark plug, the fuel spray in a spray guided system remains relatively unaffected by in-cylinder air motion but is sensitive to variations in fuel spray characteristics such as cone angle, and spray symmetry caused by manufacturing tolerances [11].

The second is wide-spacing where a large distance exists between the spark plug and the injector tip [11]. In most systems, the spark plug remains in the center of the cylinder for the reasons mentioned earlier, while the injector is placed at or near the edge of the cylinder. The wide-spacing system can achieve charge stratification in two ways. The first is by an air-guided system where in-cylinder air flows induced through swirl, tumble or squish are used to guide the fuel spray to the ignition source. Swirl is spiraling motion of air flow that has its axis of rotation parallel to the cylinder axis, tumble is a rolling motion of air flow with its axis of rotation perpendicular to the cylinder axis, and squish is the inward radial motion of air as the distance between the cylinder head and piston decreases. The second method of charge stratification is wall-guided where the fuel is injected towards a surface that directs the fuel spray to the spark plug. The surface may be a crown, bowl, or dome on a piston or the combustion chamber wall itself. In either case, for charge stratification the wide-spacing system is less sensitive to fuel spray variations than the narrow-spacing spray guided system, but more sensitive to in-cylinder air motion.

3.2.2 Charge Stratification Methods

As already briefly mentioned before, the three methods of achieving a stratified charge in a GDI system are spray-guided, wall-guided, and air-guided. The spray-guided system uses the narrow-spacing concept while the wall and air guided systems use the wide-spacing concept.

The spray-guided system is capable of achieving high charge stratification near the spark plug since the fuel spray has little time to interact with air. Therefore, the location of the spark plug relative to the injector and the fuel spray are two critical parameters that must be carefully selected in order for stratified combustion to occur. The spark plug must be located far enough from the injector to allow sufficient time for air entrainment into the fuel spray so an ignitable condition exists when the fuel spray reaches the spark plug. However, if the spark plug is too far away, over-mixing of the fuel spray and air will create a lean zone around the spark plug - resulting in a non-ignitable mixture. The location of the spark plug relative to the fuel spray is also critical as the spark plug electrode must extend into the fuel spray periphery, but not over- or under-penetrate the spray. If the plug electrode does not protrude into the spray, there will be no chance of ignition, as there is no fuel present in the spark plug gap. Likewise, if the plug electrode extends into the inner cone of the fuel spray, an overly lean condition at the electrode will also prevent ignition.

The spray guided system is also very sensitive to variations in fuel spray such as cone angle, spray symmetry, and droplet size. Changes in spray cone angle will affect the location of the spark plug relative to the periphery of the fuel spray. A decrease in the spray cone angle will effectively move the spark plug out of the fuel spray periphery while an increase in the spray angle will move the plug to the inside of the spray cone. In either case, the plug will not be in a zone ignitable mixture. Poor spray symmetry creates an unbalanced fuel mass distribution in the cylinder and, depending on the orientation of the fuel spray in-balance to the spark plug, the plug electrode could be in a zone of rich, lean, or ignitable fuel mixture. Variations in droplet size also affect the performance of a spray guided system. An increase in droplet size would require a longer distance between the spark plug and the injector since the larger droplets require longer time to evaporate. Likewise, a decrease in droplet size will require a shorter distance between the injector and the spark plug.

While the spray-guided system is sensitive to variations in fuel spray characteristics, it is less sensitive to in-cylinder air motion than an air-guided or wall-guided system when being operated in a stratified condition. With the close location of the spark plug and the injector, a small time exists between the injection event and the ignition event. This time is short enough that in-cylinder flow fields do not appreciably affect the fuel spray before the ignition event occurs. However, squish action occurring in the ten degrees before and after top dead center should affect the combustion process during stratified combustion.

The second method of charge stratification in a GDI system is the air-guided combustion system. This system usually locates the spark plug in the center of the chamber and injector at the edge of the cylinder. The air-guided system relies on in-cylinder air motion to achieve charge stratification and is thus highly dependent on in-cylinder flow fields. Because of this, the robustness of stratified combustion degrades quickly with variations in the in-cylinder flow fields. Additionally, there is a small range of acceptable calibration parameters that provide stable and efficient stratified operation. This is because the piston location is continuously changing affecting the in-cylinder flow fields. Therefore the injection timing in an air-guided system needs to be optimized for different engine speeds and load conditions, more so than in a spray-guided system. However, for homogeneous operation in a four-stroke engine with the injector located at the edge of the cylinder below the intake valves, the use of incoming air on the intake stroke greatly helps with fuel homogenizing and dispersion through out the cylinder.

The third method of charge stratification is the wall-guided system. This system is not too dissimilar from the air-guided system as the sensitivity of stratified combustion is less dependent on fuel variations and more dependent on in-cylinder flow. Wall-guided charge stratification also uses a centrally mounted spark plug with the injector located at the edge of the cylinder, but charge stratification is achieved by injecting the fuel spray towards a surface which redirects the spray to the spark plug. Hydrocarbon emissions usually increase in a wall guided system due to wall wetting caused by fuel impingement onto the wall surface. Proper injection timing is also crucial to successful stratified operation as the piston location, piston speed, and in-cylinder flows have a large impact on guiding the fuel to the spark plug. A wall-guided system shares the same benefits for homogeneous operation in a four-stroke as an air-guided system due to the placement of the injector at the periphery of the cylinder.

3.2.3 Injector Type

Currently there are two commonly accepted types of GDI injectors, with the differences between the two being how the fuel is injected into the engine. The first and more common type of GDI injector is the single fluid injector. This injector injects only fuel at a high pressure into the combustion chamber. The fuel pressure is developed either externally, with a high pressure fuel pump that transfers fuel to the injectors via a common rail fuel delivery system, or internally through the water hammer principle where a moving column of fluid that is suddenly stopped creates a sudden spike in pressure.

The second and less common type of GDI injector is the Pulse Pressurized Air Assisted (PPAA) injector. This type of injector uses a much lower fuel supply pressure than the single fluid injector but relies on pressurized air to aid in the injection of the fuel into the cylinder. A high pressure air pump delivers pressurized air to each injector and the flow of air is controlled at the injector usually by a second solenoid separate from the solenoid used to control the opening and closing of the nozzle needle. The fuel is kept in a cavity in the injector at the line pressure that it was delivered at. Upon start of injection, the air solenoid opens, pressurizing the fuel cavity, and then the fuel solenoid opens, allowing the fuel and air to flow into the combustion chamber.

Both types of injectors can use either an inwardly or outwardly opening needle in combination with a host of various nozzle designs that allow the tailoring of a fuel spray to a specific application. The outwardly opening nozzle produces a hollow cone fuel spray with a sheet thickness that is directly related to the stroke of the needle. Thus, changes in the needle stroke can be made to tailor the sheet thickness to a specific application. However, this nozzle is not usually used in combination with nozzles that change the shape of the fuel spray from that of a cone. The inwardly opening nozzle is capable of providing both a hollow and solid cone of fuel spray, with the sheet thickness of the hollow cone being proportional to the amount of swirl induced into the fuel spray as it passes through the needle/seat area of the injector as opposed to being controlled by the stroke of the needle. Additionally, the inwardly opening nozzle is more readily adapted to different nozzle designs capable of producing differently shaped fuel sprays. Such nozzles include the slit, multi-hole, shaped, and L-type nozzles.

3.3 UNIVERSITY OF IDAHO DESIGN

The type of GDI system used by the University of Idaho is narrow-spaced, spray guided and uses an outwardly opening injector that produces a hollow cone fuel spray adapted from the Evinrude E-Tec line of outboard motors. The fuel injector is a high pressure, single fluid unit injector capable of injecting fuel at 550-600 psi with 35 psi of supply fuel pressure. The fuel pressure is generated internally through the use of a moving voice coil that actuates a poppet valved plunger. The voice coil is actuated by passing current through the windings in the coil, generating a magnetic field that reacts with the field produced by a set of rare earth magnets. The magnitude of the forward driving force exerted on the plunger, along with the diameter of the plunger determine the pressure developed in the injector. The stroke of the plunger, which is controlled by the duration of the current provided to the coil, determines the quantity of fuel injected. Thus the E-Tec injector is a volume injector and the mass of the fuel injected is sensitive to variations in fuel density. Figure 10 is a diagram of the E-Tec injector [Courtesy of BRP].

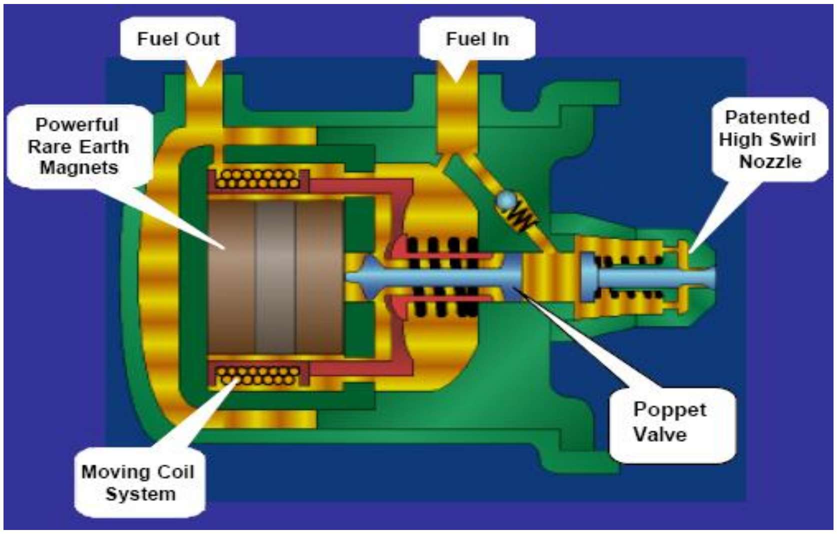

Figure 10: *Cross Section of the E-Tec injector [Courtesy of BRP].*

The nozzle used in the E-Tec injector is an outwardly opening design that produces a hollow cone of fuel. This type of fuel spray is sensitive to variations in both fuel temperature and cylinder pressure. An increase in fuel temperature or an increase in the ambient pressure into which the fuel spray is injected will cause the fuel spray to collapse to a smaller cone angle. The nozzle used in the E-Tec injector also incorporates a high swirl design (Figure 11 [3]).

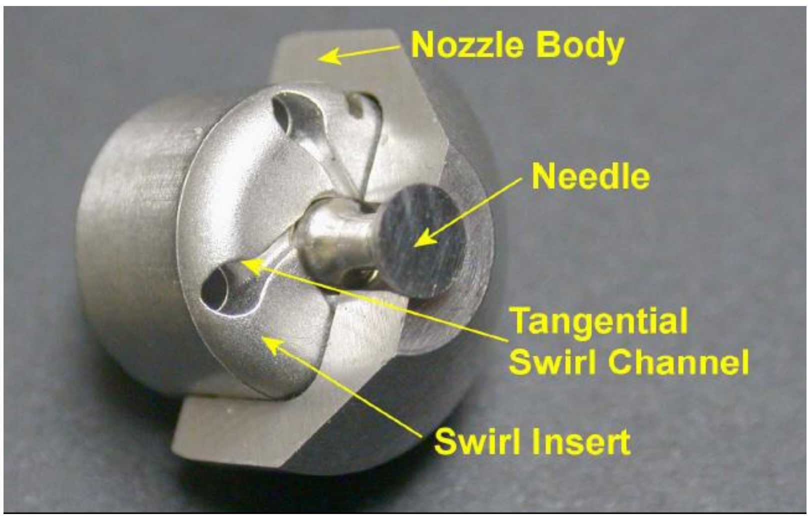

Figure 11: *Outwardly opening high swirl nozzle used in the E-TEC injectors [3].*

Fuel enters the rear of the nozzle body and flows through the tangential swirl channels in the swirl insert. This gives the fuel a rotational velocity component in addition to the linear component parallel to the needle. The rotational component improves air entrainment into the fuel spray [11]. Figure 12 shows air entraining into the fuel spray, which is seen in the image as the swirling/tumbling of fuel spray on the sides of the main fuel spray body [Courtesy of BRP]. The image on the left is fuel spray into ambient pressure while the spray on the right is into pressurized air.

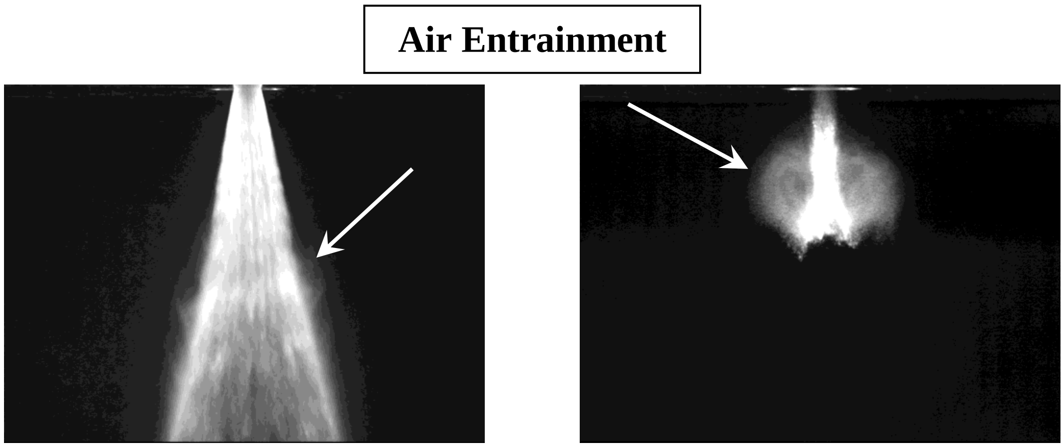

Figure 12: *Air entrainment into fuel spray [Courtesy of BRP].*

In the UIGDI design, the spark plug is placed close to the injector nozzle and is directly wetted by the fuel spray during both stratified and homogeneous operation. The combustion chamber is taller than traditional carbureted chambers, and offset to the exhaust side of the cylinder. The increased height of the chamber reduces fuel impingement onto the piston top during stratified operation at idle. Offsetting the chamber to the exhaust side of the cylinder centralizes the fuel spray which is targeted to the intake ports for increased fuel trapping during homogeneous operation [3, 10]. Figure 13 [10] shows two possible fuel spray targeting strategies with “B” being determined as the better method for the Schnurle loop scavenged engine (Section 1.5) during homogeneous operation [10].

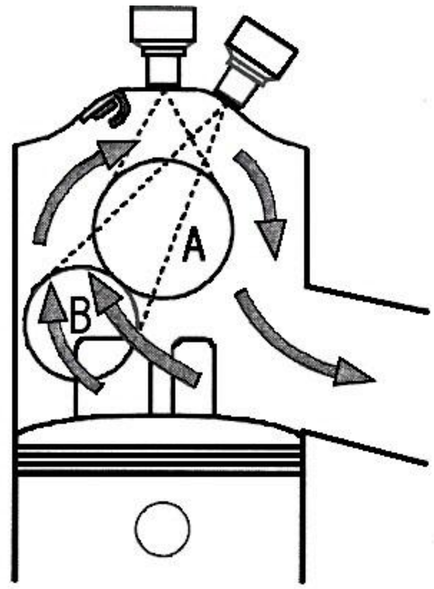

Figure 13: *Different targeting strategies for a Schnurle Loop scavenged engine [10]*.

Targeting the fuel spray to zone "B" was done to improve fuel mixing and air utilization during homogeneous operation where penetration of the fuel deep into the cylinder is necessary.

A cross section of the resulting design is shown in Figure 14 [12]. As can be seen, the main part of the combustion chamber is shaped like a cone with an abrupt transition at its base from the squish region into the main part of the chamber. The injector is located at the top of the chamber and is tilted eleven degrees to target the fuel spray towards the intake ports in the cylinder.

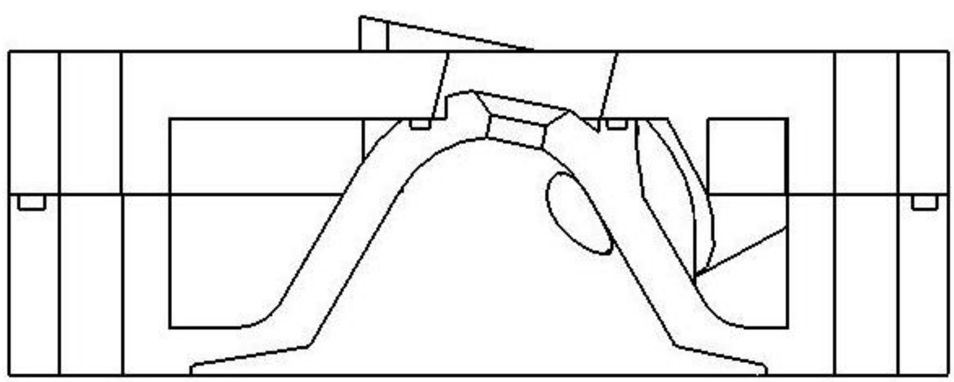

Figure 14: *Combustion chamber shape used during testing [12]*.

The spark plug hole can be seen on the exhaust side of the cylinder head and its location is such that the spark plug is directly wetted by the fuel spray, thus allowing for both stratified and homogeneous operation. The height of the chamber was calculated to limit fuel impingement onto the piston and to achieve a proper compression ratio. Figure 15 [9] shows the relationship between the fuel spray and spark plug along with the relationship between the injector tip and spark plug.

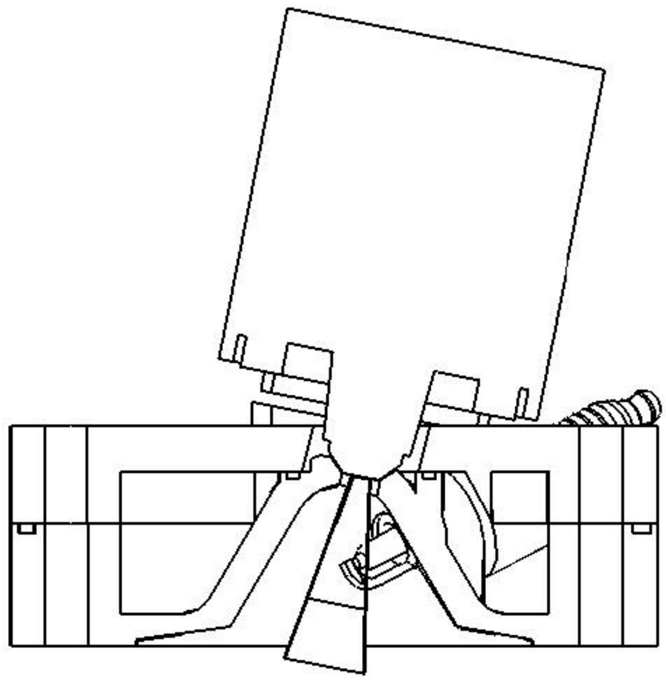

Figure 15: *Fuel spray and spark plug relationship on the UI GDI engine [9].*

4.0 SPRAY-GUIDED GDI COMBUSTION

4.1 CYCLE TO CYCLE VARIATIONS

Cycle-to-cycle variability in cylinder pressure produced during combustion is a prominent problem in two-stroke engines that directly affects run quality, emissions, and fuel consumption. The variations in cylinder pressure are directly linked to the combustion process, since the amount of heat released during combustion determines the peak cylinder pressure for any given combustion event. Heywood [8] discusses several factors that affect the combustion process in a two-stroke engine. These factors are outlined in Table 5.

Spark energy dishcarged into air-fuel mixture (1)
Flame kernel motion (1,3)
Heat losses from kernel to spark plug (1)
Local turbulence characteristics near plug (1)
Local mixture composition near plug (1)
Overall charge composition -air, fuel, EGR (2,3)
Average turbulence in the combustion chamber (2,3)
Large-scale features of the in-cylinder flow (3)
Flame geometry interaction with the combustion chamber (3)

(1) Affects early stages of combustion
(2) Affects main flame propagation process
(3) Affects the later stages of combustion.

Table 5: *Causes in cycle-to cycle variations of cylinder pressure [8].*

In addition to these factors, the combustion process of a spray-guided GDI two-stroke is further affected by the cycle-to-cycle variations in fuel spray, especially during stratified operation where pre-combustion cylinder pressures drastically affect the shape and penetration velocity of the fuel spray.

4.2 FLAME KERNEL DEVELOPMENT

The stable development of a flame front is important to ensuring stable combustion with low misfire rates. The combustion flame starts with the development of a flame kernel in the spark plug electrode gap. During the flame kernel development stage of the combustion process, there is fluid motion between the spark plug electrode and grounding strap. As the spark discharges from the electrode, a flame kernel develops and is carried away from the spark plug due to in-cylinder fluid motion. The type of ignition system used will affect the range of conditions for which a flame kernel will develop. In a capacitive discharge ignition (CDI) system, multiple sparks are used to ignite the fuel-air mixture in an attempt to increase the chances of discharging energy into an ignitable fuel-air mixture. The sparks are high relatively high in voltage, but intermittent. In an inductive ignition, energy is discharged into the fuel-air mixture continuously over a period of time. Once a spark has been established across the gap of the plug, the spark is sustained continually over a certain duration time. The inductive ignition allows a continual discharge of energy into the fuel-air mixture as it passes through the spark plug gap. Thus as a flame kernel develops and leaves the gap, the continual supply of energy to the fuel-air mixture continues the growth of the flame kernel into a developed flame capable of sustaining itself and growing. With a CDI system the growth of the kernel into a flame is done once the kernel has left the spark gap without the continual addition of energy, thus the kernel must develop into a flame on its own. So for a stratified mode of combustion where fluid motion through spark plug gap is high, the use of an inductive ignition should increase the robustness of flame kernel development, improving combustion robustness and stability. This is especially true for high load stratified operation as the late injection/early ignition strategy of engine calibration limits the amount of time available for entrainment of air into the fuel spray. This causes variations in air-fuel ratios from extreme lean at the outer boundaries of the combustion chamber to stoichiometric/ignitable at the boundary of the fuel cone spray to extreme rich inside the fuel cone wall.

4.3 HOMOGENEOUS COMBUSTION

Homogeneous combustion occurs when the entire trapped charge of the cylinder is composed of a thoroughly homogenized mixture of fresh air, fuel, and any EGR remaining from the previous cycle. This mode of combustion is typically used during moderate to heavy load conditions such as cruise and WOT. A homogeneous or semi-homogeneous mixture can be achieved by injecting the fuel early in the cycle (150-220 degrees BTDC) and timing ignition to occur late in the cycle (12-30 degrees BTDC). During these injection timings, the cylinder pressure is close to atmospheric, which allows the fuel spray to be a fully developed cone with a maximum penetration velocity dependent on droplet size and in cylinder flow fields. For a GDI system to produce maximum WOT power, the fuel spray must penetrate deep into the cylinder to maximize fuel vaporization, mixture preparation, and air utilization. This requires larger droplet sizes which produce a high fuel spray momentum capable of penetrating the intense scavenging flows that occur under high load [13]. If the droplet size is not large enough, vaporization rates will increase and the in-cylinder scavenging flows will carry fuel vapor out the exhaust port during the scavenging process, leading to increased HC emissions [13].

The location of the spark plug is also important to the power and emissions output during homogeneous combustion. The spark plug should be placed in a fuel rich area of the cylinder just before ignition occurs to ensure an ignitable mixture and complete combustion of the fuel air mixture. Homogeneous combustion is a necessity for maximum power output and moderate engine loads as air utilization is high. However the down side is that the quantity of fuel injected must be enough to create a global, ignitable mixture in the entire cylinder. This requirement leads to increased fuel consumption and higher emissions during light load conditions.

4.4 STRATIFIED COMBUSTION

Stratified combustion occurs when fuel is injected late in the cycle (0-90 degrees BTDC) creating a non-homogeneous mixture of air, fuel and EGR. Due to the late injection timing, the injected fuel remains close to the spark plug and does not penetrate the cylinder as far as in homogeneous combustion. This creates a stratified mixture of fuel and air with a high air-fuel ratio at the spark plug electrode. The result is a local ignitable mixture near the spark plug with a globally lean air-fuel mixture in the rest of the cylinder. This allows for the injection of the minimal amount of fuel required to achieve a power output or engine speed. For this reason, stratified combustion is usually used for idle, low engine loads and speeds, and transition from these points to higher loads and speeds.

During stratified combustion, the ignition event is timed earlier in the cycle (30-50 degrees BTDC) than homogeneous combustion. This is done to start combustion as the fuel is passing the spark plug gap. Once a flame is initiated, it burns from the outer region of the fuel spray to the inner region following the flammable limits of combustion as the fuel spray mixes with the air. In other words, the vaporization process takes place as combustion proceeds through the chamber [14]. The vaporization process is the primary limiting factor for high load, high speed stratified operation, during which fuel quantities and engine speeds are not conducive to complete fuel vaporization, leading to less efficient combustion.

Since the fuel injection event is timed late in the cycle, the in-cylinder pressure is substantially higher than atmospheric, causing a reduction in the maximum penetration distance and also causing the fuel spray cone to collapse into a plume as shown in Figure 16 on the next page [Courtesy of BRP]. The collapse of the fuel spray into a plume is a critical factor, as the run quality of the engine during stratified combustion is highly dependent upon the location of the spark plug electrode gap relative to the fuel spray cone. As mentioned earlier, the spark plug electrode must protrude into the periphery of the fuel plume where there is an ignitable mixture but not extend far enough into the plume that the spark plug is in fuel rich region where combustion cannot occur. Consideration must also be given to the cycle to cycle variations in fuel spray as this directly affects the relative placement of the spark plug to the fuel plume.

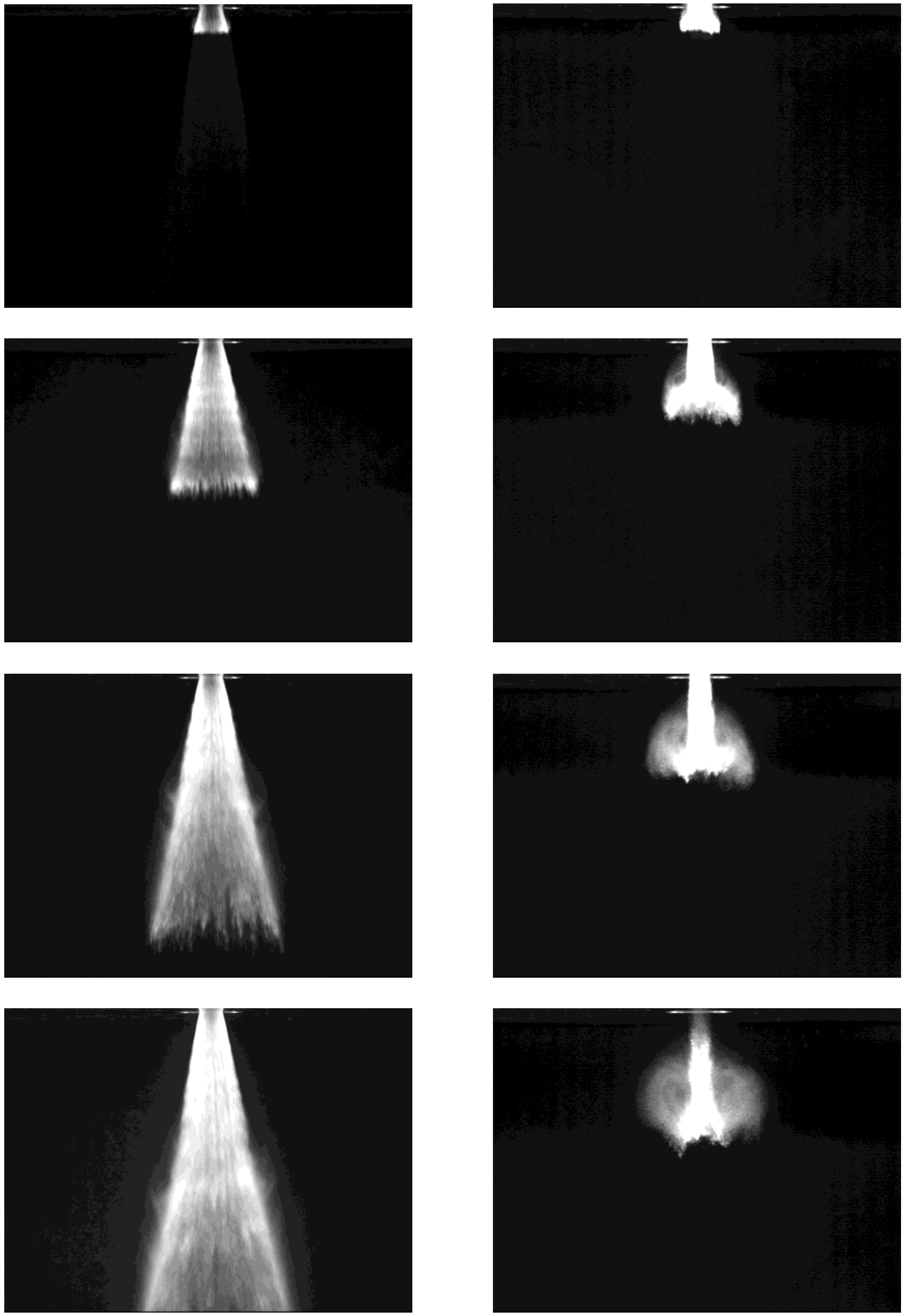

Figure 16: *Comparison of spray characteristics into ambient and elevated pressure. Images on the left are into ambient. Photos are taken 0.3 msec apart. [Courtesy of BRP]*

Stratified combustion allows the engine to run at very lean global air-fuel ratios, which improves fuel economy and UHC and CO emissions, but poses the risk of increasing NO_X emissions due to the presence of lean combustion at the flame front boundaries. Care must be given to ensure that the low levels of NO_X emissions produced by typical two-stroke engines are maintained with the adaptation of GDI systems. In addition, stratified GDI combustion is operable only at low engine speeds and loads and becomes less efficient at high loads and speeds due to reduced mixture preparation caused by high injection quantities and the shorter time available for fuel vaporization.

During homogeneous operation, the early injection timing allows for a homogeneous mixture to form throughout the cylinder. Lambda values during this mode of combustion range from rich values of ~0.9 during WOT to ~1.15-1.2 during light load conditions. Lambda is the ratio of the actual air-fuel ratio, on a mass basis, (AFR) to the stoichiometric AFR, as shown in equation 4.4.1.

$$\lambda = \frac{AFR_{Measured}}{AFR_{Stoichiometric}} \qquad 4.4.1$$

Operating the engine fuel rich during WOT ensures maximum torque and power output and also aids in piston cooling. During part load conditions, the engine is operated lean to improve fuel economy and emissions. During stratified operation, the late injection timing keeps the fuel in the upper region of the combustion chamber. Thus, only the area around the spark plug is operated at a lambda value of ~1, resulting in a globally lean condition throughout the cylinder. Because of this, the measured lambda in the exhaust stream can range from ~1.8 during high load stratified to >8 during light load and idle due to the excess oxygen from the rest of the cylinder that was not combusted.

The difference between stratified and homogeneous combustion is shown in Figure 17 (next page) [9], which illustrates the difference in lambda between stratified and homogeneous operation.

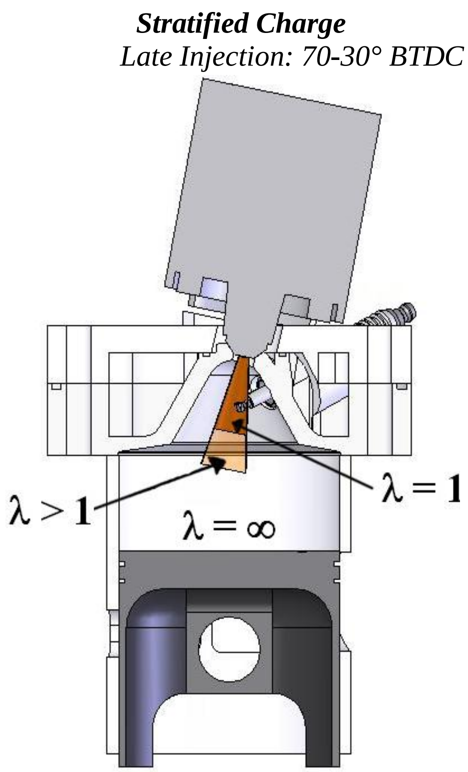

Figure 17: *Schematic showing the difference between stratified and homogeneous equivalence ratios and charge stratification [9].*

Based on the discussions presented in sections 4.3 and 4.4, one can imagine that there should be a point in the engine operating conditions where stratified combustion and homogeneous combustion are equally efficient. During calibration sessions in 2004 and 2005, this was known to exist, however not much time was spent with it as the other areas of the map needed attention. Therefore, it was decided to select a transition point that provided equal torque at homogeneous and stratified combustion. This point occurs at 30% throttle between 3000 and 6000 rpm, as shown in Table 6 [9].

Throttle Position	Engine Speed										
	500	1000	1500	2000	3000	4000	5000	5500	6000	7000	8000
10											
50			Stratified								
80											
100											
150											
200											
250											
300					Transition						
301											
400											
500						Homogeneous					
600											
700											
800											
1000											

Table 6: *Table showing engine control strategy and transition from stratified to homogeneous [9].*

Now that initial calibration procedures are completed, research can be conducted on the transition from stratified to homogeneous combustion and improving stratified combustion. These are two of the goals of this research: a) to investigate both homogeneous and stratified combustion to find the point of equal efficiency and to determine which mode of combustion is better for cruise point operation in a snowmobile, and b) investigate how squish velocity affects stratified operation both in the maximum achievable torque and in the range of efficient stratified combustion.

4.5 IMPROVEMENT OF STRATIFIED COMBUSTION

Stratified combustion allows for lower fuel consumption and reduced emissions at light load due to small injection quantities. However, because of late injection and early ignition, mixture preparation time is significantly reduced over homogeneous combustion. Thus it is desirable to have a fuel spray with small droplets in order to achieve quick fuel vaporization, which aids in ensuring the presence of an ignitable mixture at the time of ignition and ensures the vaporization of the injected fuel before the combustion front reaches it. However, at higher engine loads the limiting factor is vaporization time [3]. This is due to the increased fuel quantities required to achieve higher torque and not enough time for complete vaporization of the fuel. Likewise, at high speed, the time available for fuel vaporization is reduced, further decreasing the efficiency of stratified operation.

The collapse of the fuel spray into a plume is another limiting factor for stratified operation as the surface area of the fuel spray decreases the surface area that is available for air entrainment. During high load stratified operation large fuel quantities are injected. This further amplifies the problem of air entrainment into the fuel spray, as the late injection angle used during stratified operation limits the time available for mixing to occur. Torque output is limited at low speeds due to the window of injection angle that still allows stratified operation to occur. At low speeds, if the injection angle is too far advanced to promote air entrainment, the fuel spray is injected into a lower density cylinder charge and the cone is better developed and has deeper penetration into the cylinder. This does two things: first, the better developed fuel cone effectively moves the spark plug location further into the fuel spray where there is less of a chance of an ignitable mixture being available. Second, the deeper penetration allows the fuel spray to over-penetrate the spark plug causing more fuel to be placed further into the cylinder leading to over-mixing during lower fuel loads and a heterogeneous fuel-air charge during high fuel loads. The heterogeneous charge is neither a homogeneous mixture nor a stratified mixture but a combination of both. It is not as efficient as a homogeneous charge and increased fuel consumption and emissions result from incomplete combustion.

During stratified operation, the amount of atomized fuel at the start of ignition is significantly less than that of homogeneous operation, decreasing the chances of an ignitable fuel-air mixture passing through the spark plug gap. This is especially true at higher engine loads during stratified operation when large amounts of fuel are being injected into the cylinder. There are several methods of increasing the atomization rate of the fuel spray. The first is to increase the temperature of the air that the fuel is being injected into. The higher in-cylinder air temperature increases the heat transfer into the fuel spray raising the temperature of the fuel and increasing the atomization rate. This can be accomplished by increasing the operating temperature of the engine, which has been found to improve stratified combustion [3]. Another method of decreasing vaporization time is to increase the temperature of the fuel. However, the E-Tec injectors inject fuel on a volume basis and thus changes in fuel temperature affect fuel density and the mass of fuel injected. In either case, the ability to control fuel temperature or engine temperature as a function of combustion mode is not currently available. Therefore these methods were not pursued.

A third method to decrease vaporization time is to decrease the Sauter Mean Diameter (SMD) of the fuel spray. A smaller SMD yields a higher surface area to volume ratio for individual fuel droplets, increasing the heat transfer rate into the droplets and allowing the temperature of the fuel drops to reach the vaporization temperature more quickly. Decreasing the SMD is primarily accomplished through higher injection pressures. However, smaller SMD values decrease the high load, high speed homogeneous combustion power output as the smaller drops have less momentum and in turn cannot penetrate the counter flow of in-cylinder air [13]. This leads to decreased air utilization, higher temperatures before ignition, higher knock tendencies at maximum brake torque (MBT) ignition values resulting in less than optimum MBT ignition values and ultimately reduced power output. However, due to hardware constraints, this method was not pursued.

A fourth method of increasing the atomization rate of the fuel is to increase the relative velocity between the fuel spray and the air into which it is injected into. Higher relative velocities tear the fuel spray apart better than lower relative velocities, resulting in improved atomization. The interaction of the injected fuel with the in-cylinder motion of the air charge breaks up the fuel spray, increasing the amount of fuel exposed to air and thus increasing fuel-air mixing and air utilization. Increasing the relative velocity between the fuel spray and in cylinder fluid motion improves the mixing of fuel and air during high fueling quantities which decreases the air-fuel ratio gradient across the combustion chamber during stratified operation.

One method of increasing the relative velocity between the fuel spray and the in-cylinder air is to increase the squish velocity. During the compression process as the piston rises towards TDC the volume between the piston top and the squish band of the cylinder head decreases, rapidly forcing the air in this region into the main part of the combustion chamber (Figure 18).

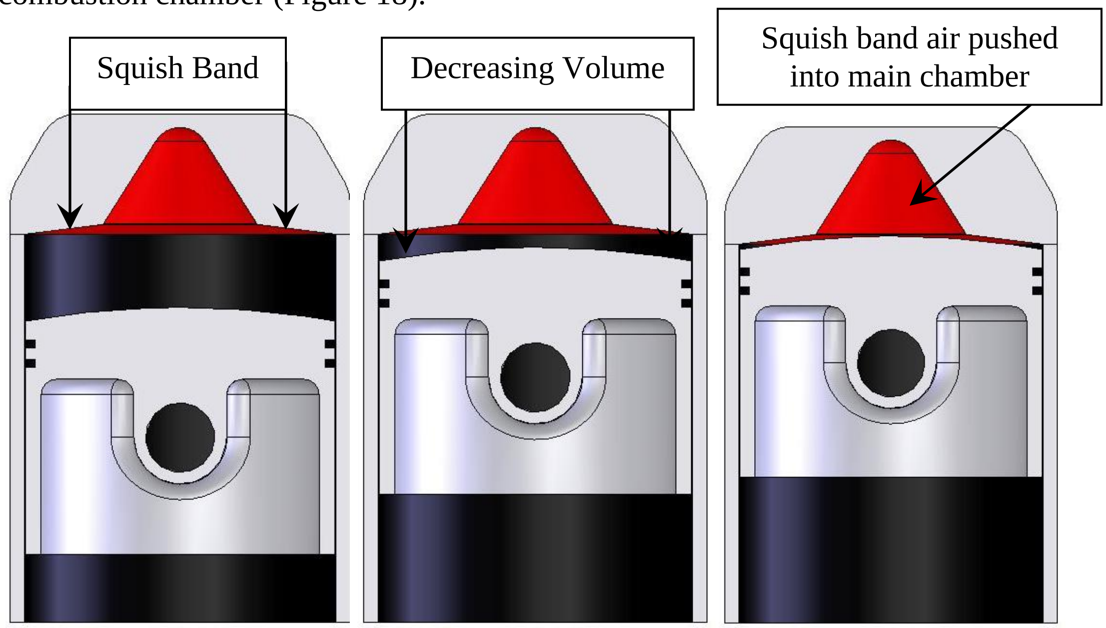

Figure 18: *Illustration of decreasing squish volume as piston approaches TDC.*

The effect of the squish band is most pronounced in the 10 degrees before and after TDC, which is the same location that stratified combustion and fuel vaporization is occurring. The peak squish velocity depends on the squish area, location of the squish region, and clearance between the piston top and squish band in the cylinder head.

Since fuel vaporization occurs during the combustion event, stratified combustion requires vigorous in-cylinder air motion to rapidly vaporize the fuel spray before the combustion front reaches it [14]. This will aid in both high speed operation and high load operation. During high speed stratified operation the effects of squish should aid in entraining air into the fuel spray, thus allowing later injection timings at high speeds, reducing fuel over-mixing and improving BSFC and emissions. During high load stratified operation when larger quantities of fuel are being injected, the effects of squish will help move air from the perimeter of the cylinder into the fuel rich zone in the center part of the combustion chamber. This should improve combustion efficiency as well as combustion robustness. In both cases, the effect of squish will be to increase the vaporization rate of the fuel so when the combustion front reaches the richest parts of the fuel plume, allowing complete combustion to occur.

Blair discusses several combustion chamber designs that are commonly found in internal combustion engines and recommends the offset, total offset, and deflector designs, as shown in Figure 19, for stratified operation in a loop or cross scavenged GDI two-stroke engine.

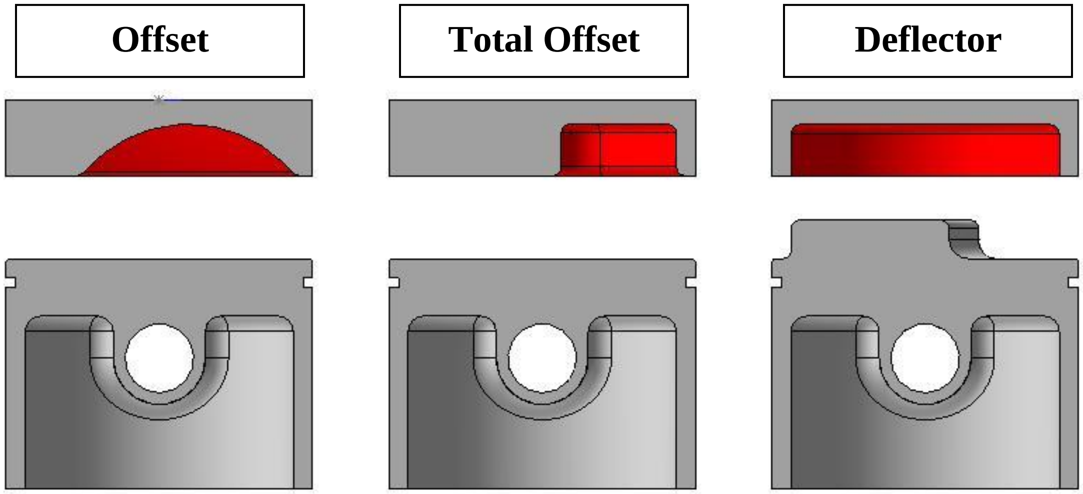

Figure 19: *Illustration of different combustion chamber designs for improved squish.*

Blair also mentions that a bowl in piston design (Figure 20) further improves fuel vaporization by inducing turbulence to the squish flow as it passes over the squish area on the piston and into the bowl of the piston near the end of compression [14].

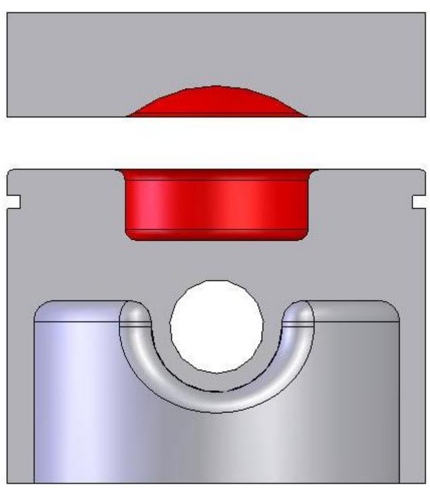

Figure 20: *Bowl-in-piston design for improved squish action for stratified operation [Adapted from 14].*

However, one drawback to increasing the squish velocity in an engine is that during high load high speed operation, the increased squish velocity increases the chance of inducing detonation. This is because as the piston forces the air-fuel mixture from the squish region into the main chamber, energy is put into the mixture. If the squish velocity becomes too high, the energy input into the fuel-air mixture will cause a large enough temperature increase to cause spontaneous ignition, or pre-ignition, leading to detonation and engine damage. Typically, squish velocities of 18-20 m/s at the rated speed of the engine are accepted as the maximum before detonation issues become a problem. However, at the speed range of stratified operation, these values may not be enough to significantly contribute to the mixing process.

The work of Szekely and Alkidas [15] shows other methods of improving stratified combustion in a spray-guided system, but due to limitations in hardware were not pursued in the research presented here. However, they are worth mentioning. These include rate of injection, spark plug electrode length, and spark energy. They found that decreasing the rate of injection by using a lower needle lift and longer injection duration improved fuel consumption by reducing the 50-90% burn duration. The decreased 50-90% burn duration was created by delaying the end of injection further into the combustion process, resulting in a richer fuel mixture towards the end of the combustion process [15]. Schanzlin, et. al. [16] found that stratified combustion was slow at the end of combustion because the flame front expands to areas of lean mixtures, decreasing the flame speed significantly. Szekely, et. al. [15] also go on to say that the optimum spark plug electrode length was one that allowed the electrode to protrude into the fuel spray. This led to reduced fuel consumption, UHC emissions, smoke (particulate) emissions, and improved combustion stability. Szekely, et. al. [15] also found that the coefficients of variation in indicated mean effective pressure (COV of IMEP) was independent of flame kernel development and dependent on the amount of fuel consumed during combustion. This showed to reduce the COV of IMEP and illustrates that it is more important to consume all of the injected fuel during the combustion process than to properly phase the combustion process. Therefore, by phasing the combustion process early in the cycle, a more complete combustion process can be ensured [15].

The combustion chamber design used during testing is an offset chamber with the center of the chamber offset 1.905 mm (0.075”) from the centerline of the cylinder toward the exhaust to center the fuel spray in the cylinder. This combustion chamber was also designed for use in a turbocharged application; therefore the combustion chamber volume is larger than normal to decrease the compression ratio of the engine. This chamber geometry was used because cylinder head was already manufactured, which is a detailed and time consuming process. Therefore, redesign of the combustion chamber was not considered as an option, nor was piston re-design, as the manufacturing of custom pistons is another process beyond the scope of this project. Thus, the only method of changing squish velocity is to change the clearance between the piston top and the squish band of the cylinder head. This was the route that was selected.

4.6 ENGINE MODIFICATION

There were two possible methods of reducing the squish clearance on the test engine. The first method would have been to remove material from the bottom of the cylinder, moving the cylinder and head closer to the crankshaft centerline, thus reducing the clearance between the piston and squish band at TDC. This approach would also change the port timing of the engine. The timing of both the exhaust and scavenging ports would have been lowered. Lowering the exhaust ports increases the trapped compression ratio of the engine and also allows for a longer expansion period. However, the location of the piston relative to the crankshaft centerline at BDC remains the same. This means that the scavenging ports would not be fully uncovered while the piston is at BDC, reducing the scavenging performance of the engine.

For these reasons it was decided to remove material from the top of the cylinder. This would allow port timing to remain unchanged from test to test. However, decreasing the squish clearance also increases the compression ratio of the engine. While this will affect engine efficiency, it is thought that most of the changes in engine performance during stratified operation are contributed to the increase in squish velocity and not due to the increased compression ratio.

There were two different squish clearances used during testing. The first was with unmodified cylinders which yielded squish clearance of 0.065”. This was the clearance used during homogeneous testing and for the first set of stratified data. Figure 21 shows the squish velocity as a function of crank angle for various engine speeds observed in testing. The code used for generating this plot can be found in Appendix A.

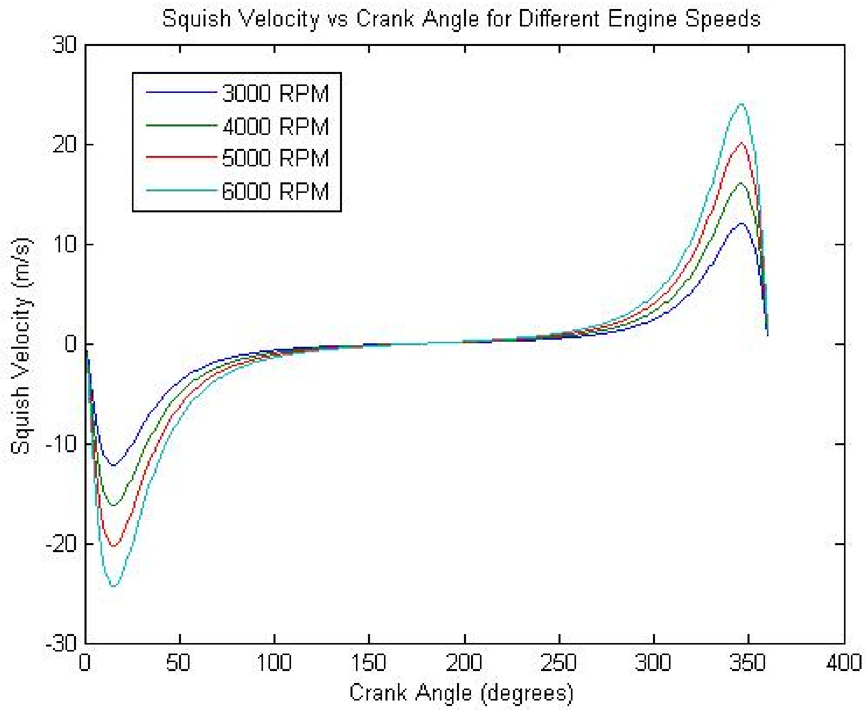

Figure 21: *Squish Velocity vs. Crank angle for 0.065" clearance at various engine speeds.*

Note that the maximum squish velocities occur ten degrees before and after top dead center, and that as engine speed increases, so does the maximum squish velocity along with the area under the velocity curve. The lowest peak is associated with 3000 rpm and the highest peak is associated with 6000 rpm. The maximum squish velocities for each engine speed and the compression ratio for both squish clearances are tabulated in Table 7.

Once the homogeneous and stratified data had been collected, the engine was disassembled and the cylinders were modified to increase squish velocity. The squish clearance was reduced 0.015" to a final clearance of 0.050" and the new squish velocity profiles are shown in Figure 22.

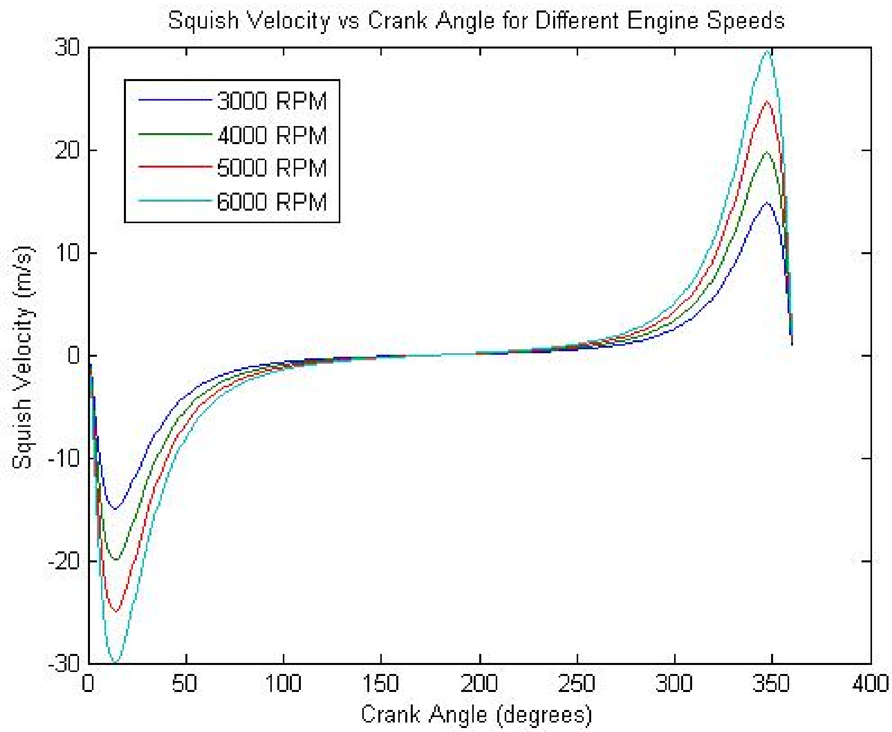

Figure 22: *Squish velocity vs. crank angle for 0.050" clearance at different engine speeds.*

Like in Figure 21, the lowest peak is associated with the 3000 rpm point and the highest peak is associated with 6000 rpm. Table 8 compares the changes in trapped and geometric compression ratios for each of the squish clearances. As can be seen, the squish values increased 22 percent at all engine speeds while the compression ratio increased two percent. It is believed that the increased squish velocity will be responsible for any changes in combustion performance during stratified operation.

Peak Squish Velocity (m/s)			**% Change**
	Clearance		
RPM	***0.065"***	***0.050"***	
3000	12.0	14.7	22.8
4000	16.0	19.7	22.9
5000	20.0	24.6	22.9
6000	24.0	29.5	22.9

Table 7: *Comparison of peak squish velocities at different engine speeds for 0.065" and 0.050" clearances.*

Compression Ratio			% Change
	Clearance		
	0.065"	***0.050"***	
Geometric	8.38	8.57	2.3
Trapped	5.92	6.05	2.2

Table 8: *Comparison of geometric and trapped compression ratios for both 0.065" and 0.050" squish clearances.*

Engine modifications were performed in the University of Idaho's Mechanical Engineering Department machine shop. After the engine was disassembled, each cylinder was measured from the cylinder head mating surface to the crankcase mating surface to check for parallelism and to see if the cylinder head surface could be squared to the mill deck through the crankcase surface. The measurements showed that both surfaces were parallel to within 0.001" and therefore the cylinder head surface could be considered square to the milling deck if the crankcase surface was square to the milling deck. Before any machining was done, the mill head was squared with the mill deck. This was done using a dial indicator attached to the mill collet and then sweeping the dial indicator in a circle across the mill deck. After adjustment, the mill head was found to be square with the mill deck to within 0.0003" in any direction.

After adjusting the mill, each cylinder was bolted to the deck as shown in Figure 23. The cylinder sat on 1-2-3 blocks and was clamped to the deck surface. A ½" diameter end mill was used to remove material from the top of the cylinder. The mill was turned on and the milling deck raised until the cutter just brushed the cylinder top. The table was then raised an additional 0.012" and a roughing pass was made. Once the roughing pass was complete, the table was raised another 0.003" and a finish pass was performed. The final amount of material removed was 0.015". Pictures of the setup (Figure 23) and machining in progress (Figure 24) and of the final product (Figure 25) are shown below.

Figure 23: *Equipment setup for machining cylinders.*

Figure 24: *Cylinder machining in progress.*

Figure 25: *Cylinder machining finished.*

This process was repeated for the second cylinder. Once machining was done, both cylinders were thoroughly cleaned and the engine was reassembled. The squish clearance was measured for verification and was found to be 0.049”.

5.0 TESTING

In order to determine which mode of combustion is superior for cruise point operation, the engine was calibrated for both homogeneous and stratified combustion over a wide range of part load operating conditions. The conditions were selected to offer a wide range of cruise point conditions and ensure that the proper location of the transition from stratified to homogeneous combustion was found. The selected range of operation included a speed range from 3000 rpm to 6500 rpm in 500 rpm increments and throttle positions of 10%, 20%, and 30% at each rpm point. Calibration of the engine occurred in increments of 10% throttle and 500 rpm and each setting will be referred to as a calibration point.

5.1 METHODOLOGY

During testing, the primary factor observed was the brake specific fuel consumption (BSFC) of the engine, which is simply the mass flow rate of fuel divided by the power output of the engine (Equation 5.1.1). Units for BSFC are either kilograms/kilo-Watt-hour (kg/kW-hr) or pounds per horsepower-hour (lb/hp-hr). BSFC was chosen because it is fairly easy to measure and it can be correlated to the arbitrary overall efficiency of the engine [17] (Equation 5.1.2).

$$BSFC = \frac{\dot{m}_f}{\dot{W}}\left(\frac{kg}{kW * hr}\right) \qquad (5.1.1)$$

$\dot{m}_f$ = mass flow rate of fuel

$\dot{W}$ = power output

$$BSFC = \frac{1}{\Delta H_0 \eta_0} \qquad (5.1.2)$$

ΔH_0 = heating value of the fuel

η_0 = arbitrary overall efficiency

Equation 5.1.2 shows that BSFC is inversely related to the arbitrary overall efficiency, which is often on the same order as thermal efficiency but is not considered the thermal efficiency of the engine [17]. Thus, lower BSFC values indicate a higher arbitrary overall efficiency. Looking at it another way, BSFC is a measure of how efficiently the engine is turning fuel energy into useable power output. A lower BSFC value means the engine is using less fuel to make the same power as a point that has a higher BSFC value.

5.1.1 HOMOGENEOUS

The first data collected were for homogeneous combustion and served as the baseline values to which all data are compared. At each calibration point the injection angle, fuel quantity and ignition timing were first set to maximize torque output. Then an injection angle sweep was performed to find the angle that produced minimum BSFC value for the current throttle setting and engine speed. The sweep was performed by retarding the injection angle from an initial value of 180-200 degrees BTDC to a minimum value of 150-170 degrees BTDC in ten degree increments, with the beginning and ending values of injection timing depending on the engine speed. During the sweep the air-fuel ratio was kept constant, thus allowing fuel quantity to change at each point. Ignition angle was also kept constant. At each injection angle torque, power, fuel consumption, ignition angle, air-fuel ratio and BSFC were recorded.

Once the injection angle with the lowest BSFC was found, fuel sweeps were performed by varying the injected fuel quantity from the rich limit of combustion (~ lambda 0.9) to the lean limit of combustion (~ lambda 1.6) in increments of 0.5mm^3 of injected shot volume. The parameters recorded during the fuel sweep were the same as in the injection angle sweep. The goal of the fuel sweep was to find a calibration setting that further reduced the BSFC beyond that of the injection angle sweep. Once a minimum BSFC value was found, the torque output at the minimum BSFC value was selected as the target torque for stratified operation. The injection angle and fuel quantity sweeps were performed for every calibration point.

5.1.2 STRATIFIED

For each calibration point during stratified operation, there were two goals. The first goal was to determine if stratified combustion could produce a lower BSFC value than homogeneous combustion while achieving the target torque set at the point of minimum BSFC during homogeneous combustion. The second goal was to determine the window of operation of stratified combustion for each calibration point in terms of injection angle and ignition delay/spark timing along with the maximum achievable torque at each rpm point for stratified combustion.

At each calibration point, an injection angle sweep was performed that ranged from the minimum injection angle that could produce the target torque to the maximum injection angle that still allowed stratified combustion. Thus the injection angles varied depending on engine speed and target torque. At each injection angle, the minimum ignition delay, maximum ignition delay, and average ignition delay were determined. This established the window of stable stratified combustion. At each delay setting, torque, power, fuel consumption, air-fuel ratio, and BSFC were recorded for comparison to homogeneous combustion and for determining the operating range of stratified combustion. This calibration procedure was repeated for each target torque at each rpm point until all calibration points were completed. Additionally, at each engine speed the maximum achievable torque was determined.

The maximum torque and minimum BSFC were recorded to determine the transition point between homogeneous and stratified combustion and to determine how the increased squish velocity affected stratified combustion. It was expected that the maximum torque output and the maximum engine speed of efficient stratified combustion would increase.

5.2 DESCRIPTION OF TESTING EQUIPMENT

5.2.1 DYNAMOMETER

The dynamometer used for testing is a nine-inch diameter water brake model specifically designed for use on snowmobile engines. Manufactured by Land & Sea, the dynamometer is composed of a dynamometer head, load valve, servo driven throttle control, water pump, water tower, and data acquisition/control software. The software collects data such as pressures, temperatures, engine speeds, dynamometer shaft speeds, torque, power, and fuel flow, just to name a few. The software also controls the engine speed and throttle setting through control of the load valve and servo driven throttle control. The dynamometer-head attaches directly to the crankshaft of the engine in place of the primary clutch, as shown in Figure 26.

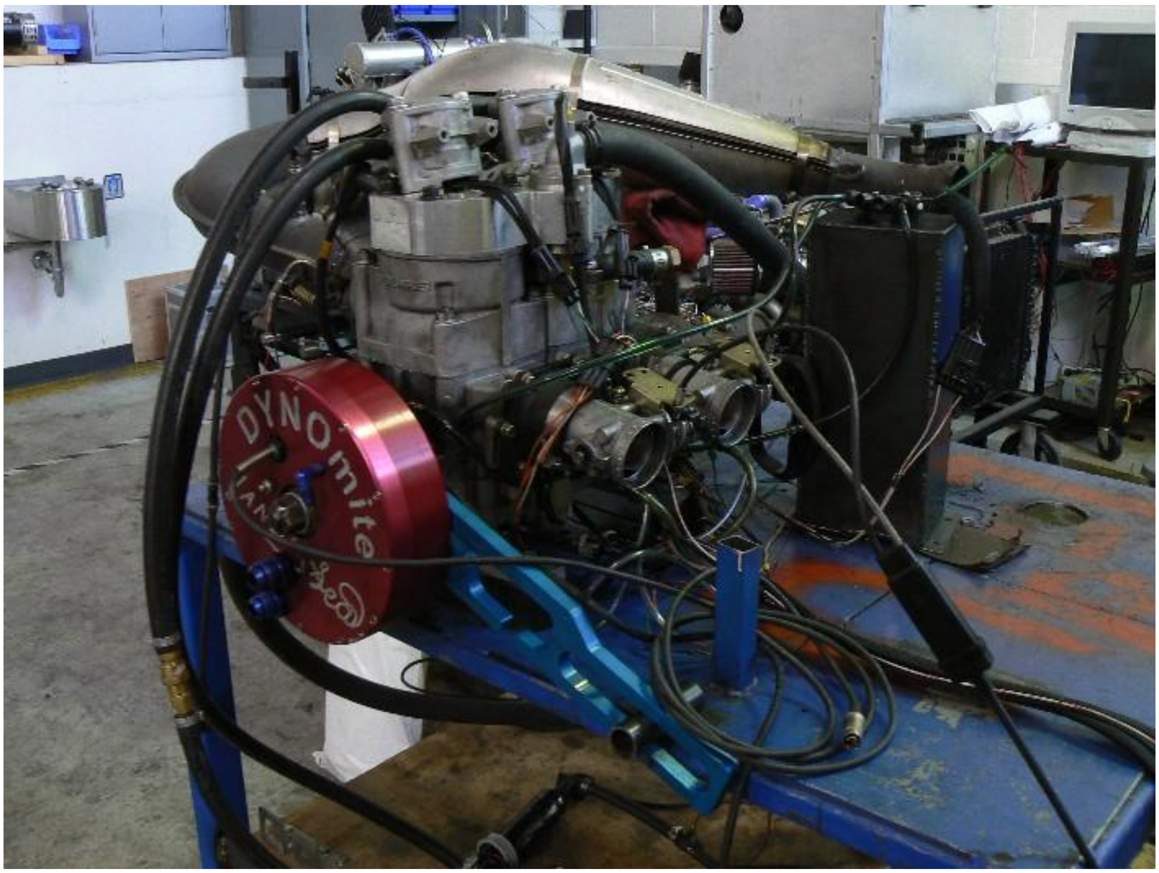

Figure 26*: Dynamometer head attachment to the crankshaft.*

The dynamometer system works by pumping water from the water tower to the load valve. Based on the settings selected by the user, the load valve is automatically adjusted to change water flow to the dynamometer-head to control engine speed, regardless of torque output. The dynamometer-head consists of a rotor and housing. The rotor is bolted to the crankshaft hub and rotates inside of the housing. The water is pumped into the rotor which then accelerates the water to the periphery of the housing, creating viscous forces causing a torque on the housing about the crankshaft high enough to match the torque output of the motor. The housing has a torque arm attached to it to prevent the housing from rotating. The arm has a full bridge strain gauge attached to it to measure arm deflection. The computer controlled load valve automatically varies water flow into the rotor, and thus the viscous forces between the rotor and housing, allowing the engine to be controlled remotely from the control room.

The full bridge strain gauge measures arm deflection and outputs a voltage to the computer, which in turn calculates torque output based on a calibration from Land and Sea. The strain gauge is temperature compensated and calibrated with a two point calibration that includes no load and a static, dead weight load. The accuracy of the strain gauge is 0.5 % of the full load range. Figure 27 shows the torque and horsepower absorbing properties of the dynamometer-head as a function of absorber speed [18].

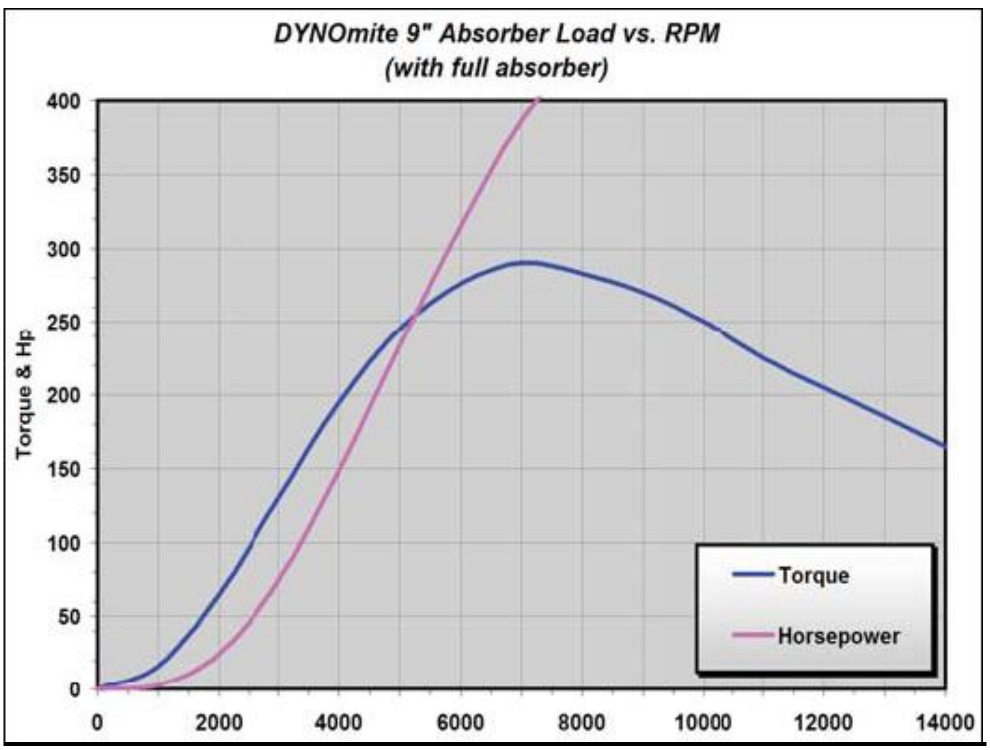

Figure 27: *Torque and Power absorbing properties of the dynamometer-head [18].*

This equipment is capable of steady state tuning where the engine speed is held constant by the computer controlled load valve that varies water flow through the dynamometer head based on torque output and desired hold speed. This allows for "cell-by-cell" engine mapping where each throttle position and engine speed cell in the engine map can be calibrated by holding a specified engine speed and varying the throttle to calibrate the engine maps. This system is finicky in the early stage of engine mapping where incomplete maps cause erratic changes in torque output. This causes the load valve to drastically change water flow to maintain engine speed, however once the engine reaches a higher torque output the engine speed increases and the load valve compensates again. This leads to a cyclic oscillation that can only be stopped by going to a lower engine speed and throttle setting where torque output is reduced. However, once a full map has been created and torque output of the engine is stable through all map cells, the equipment functions normally but deviations in the desired hold speed can still be as high as 100 rpm.

5.2.2 FUEL MEASUREMENTS

Fuel measurements were taken using a Max Machinery 710 Series Fuel Measurement system shown in Figure 28. This is a two piece unit consisting of a fuel conditioning/metering package and a display unit.

Figure 28: *Fuel metering and display units.*

The metering unit contains the fuel conditioner, fuel pumps, and metering device. It measures fuel flow on a volume basis and then internally calculates and displays the mass flow based on a user defined calibration curve that uses the density of the fluid to calculate mass flow to within an error of 3%. The unit is equipped with a thermocouple to adjust the calibration based on temperature changes in the fuel. The meter is capable of storing up to four different calibration curves for use with different fuels. The unit can supply 5 psig for carbureted applications up to 100 psig for fuel injected applications. During testing, the stock engine fuel pump was used to generate the pressurized fuel supply to the injectors while the fuel cart supplied 5-10 psig of fuel to the stock engine fuel pump.

The display unit shows mass flow in real time, fuel temperature, and is also capable of recording mass of fuel flowed over a period of time for time averaging fuel flows. During testing this feature was used to take 15 second averages of the real time mass flow being displayed. This helped to smooth out drift in the measuring system and proved to drastically improve the accuracy and repeatability of readings.

6.0 RESULTS AND CONCLUSIONS

Following are the results of homogeneous, stratified, and high squish stratified combustion testing. The results are presented in tabulated form for easy comparison of individual throttle and rpm cells.

6.1 HOMOGENEOUS

Using the procedures outlined in section 5.1.1, each calibration point resulted in a set of data that showed trends in BSFC relative to injection angle and then to fuel quantity or essentially Lambda. The lambda sweeps were performed at the injection angle that produced minimum BSFC during the injection angle sweep. Figure 29 shows the dependency of BSFC on injection angle while the air fuel ratio is held constant. The air fuel ratio was held constant so that torque and power output of the engine remained the same during the injection angle sweep.

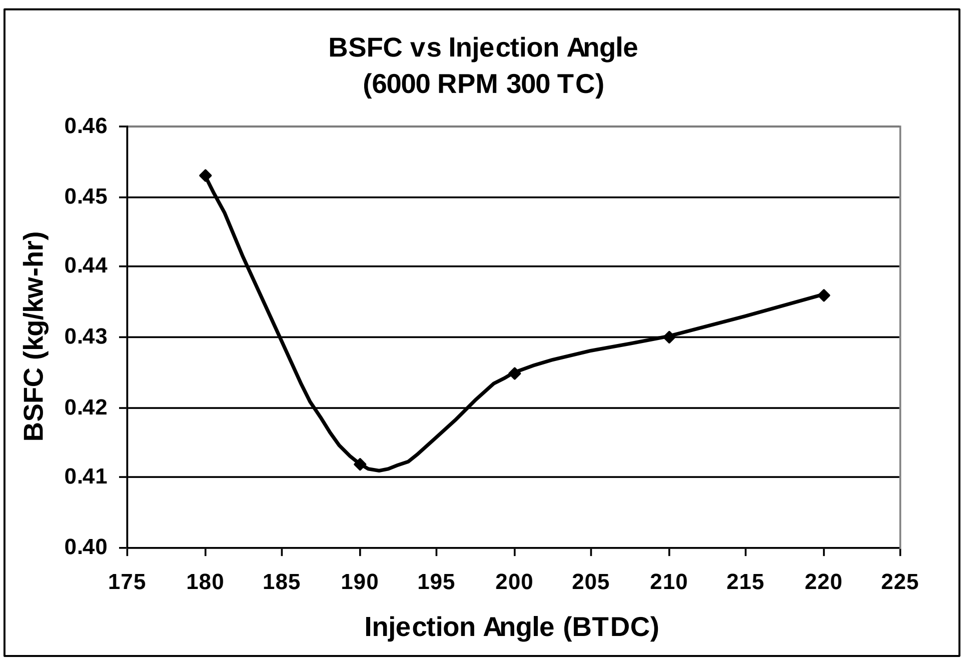

Figure 29: *BSFC vs. Injection angle for 6000 rpm and 30% throttle with lambda = 1*

These data are for 6000 rpm and 300 TC during homogeneous combustion. The variations in BSFC are due to differing commanded shot volumes required to maintain a constant air fuel ratio. At the earlier injection timings (higher injection angle) the amount of short circuited fuel is higher than at the later injection timings (lower injection angles). This results in higher commanded shot volumes and higher fuel flow rates to maintain a constant lambda. Figure 29 shows that as injection angle decreases from 220 degrees BTDC to 190 degrees BTDC, so does the BSFC. This is because the power output of the engine is remaining constant, but the later injection timing reduces fuel short circuiting, decreasing the fuel flow required to maintain a constant lambda value. At an injection angle less than 190 degrees BTDC, the BSFC increases. This is due to injection timing that is too late, which decreases mixing time for the fuel and air. This reduces air utilization and results in reduced power output. Based on this data at this calibration point, the 190 degree injection angle was chosen as the best injection angle to perform a fuel/lambda sweep. Figure 30 shows how the air fuel ratio that the engine is operated at affects BSFC.

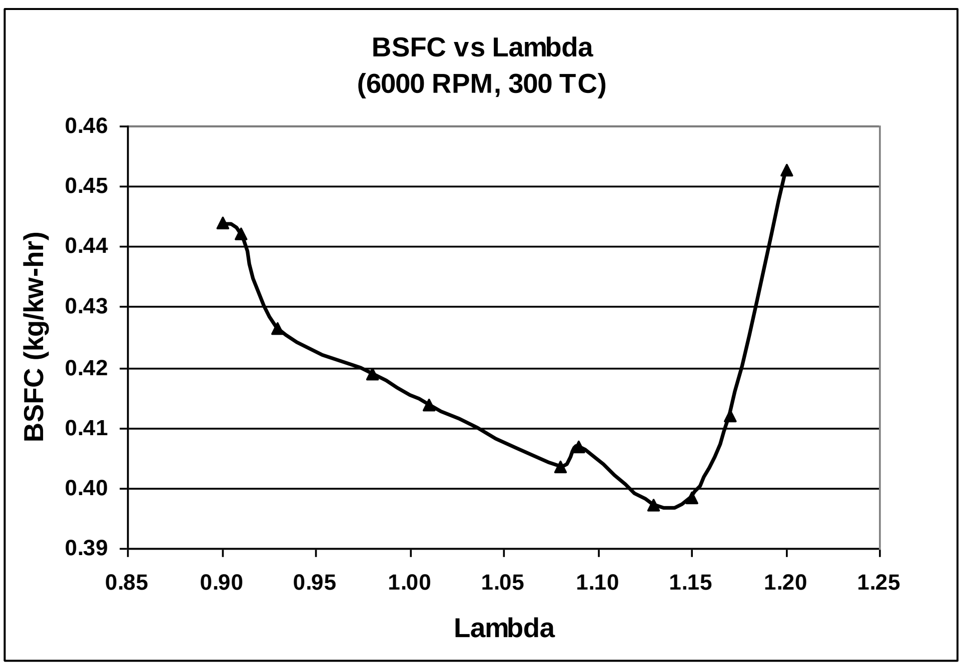

Figure 30: *BSFC vs. Lambda Injection angle was held constant at 190 DBTDC*

Again, lambda is the ratio of the measured air-fuel ratio to the stoichiometric air-fuel ratio. A value less than 1.0 is fuel rich while a value greater than 1.0 is fuel lean. Note that the stoichiometric value of 1.0 does not produce the minimum BSFC value. Instead the minimum BSFC value occurs just slightly lean of stoichiometric, with changes in either direction (leaner or richer) increasing BSFC. Thus, the engine is more efficient at converting fuel energy into power at leaner air-fuel ratios.

This is because BSFC is a balance of power produced and fuel flow required to achieve that power. The power output of an engine is directly related to the air-fuel ratio at which it operates with richer air-fuel ratios providing more power but at a higher fuel flow rate leaner air-fuel ratios producing less power but with lower fuel flow rates. Figure 31 shows power output for 6000 rpm and 30% throttle as a function of lambda.

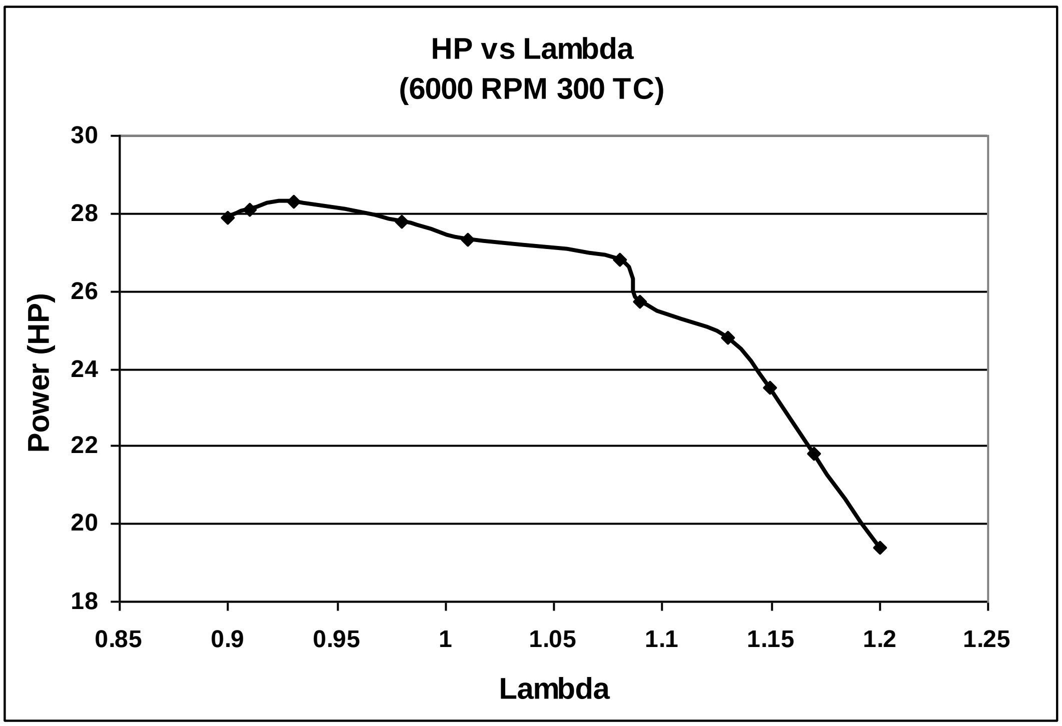

Figure 31: *Power vs. Equivalence ratio for 6000 rpm and 30% throttle. Note that operating the engine fuel rich produces more power to a certain point.*

Note that as the engine is operated under richer conditions, it produces more power until a certain point. At this point, the rich limit of combustion is reached and misfire rate increases, causing poor run quality and reduced power output. Likewise, as lambda increases, there is a decrease in the power output of the engine. This is due to smaller quantities of fuel being burned which leads to lower amounts of heat released during combustion. Beyond a lambda of 1.2, the lean limit of combustion is reached leading to intermittent misfire, reducing the run quality of the engine and power output.

Looking at the fuel flow as a function of lambda at 6000 rpm and 30% throttle, we see that there is a direct relationship between the two. (Figure 32) This makes sense since the air fuel ratio is calculated based on the ratio of the mass of air and fuel combusted. Since the mass air flow rate through the engine is held constant, any changes in the fuel flow rate through the engine should be a direct, linear relationship to lambda.

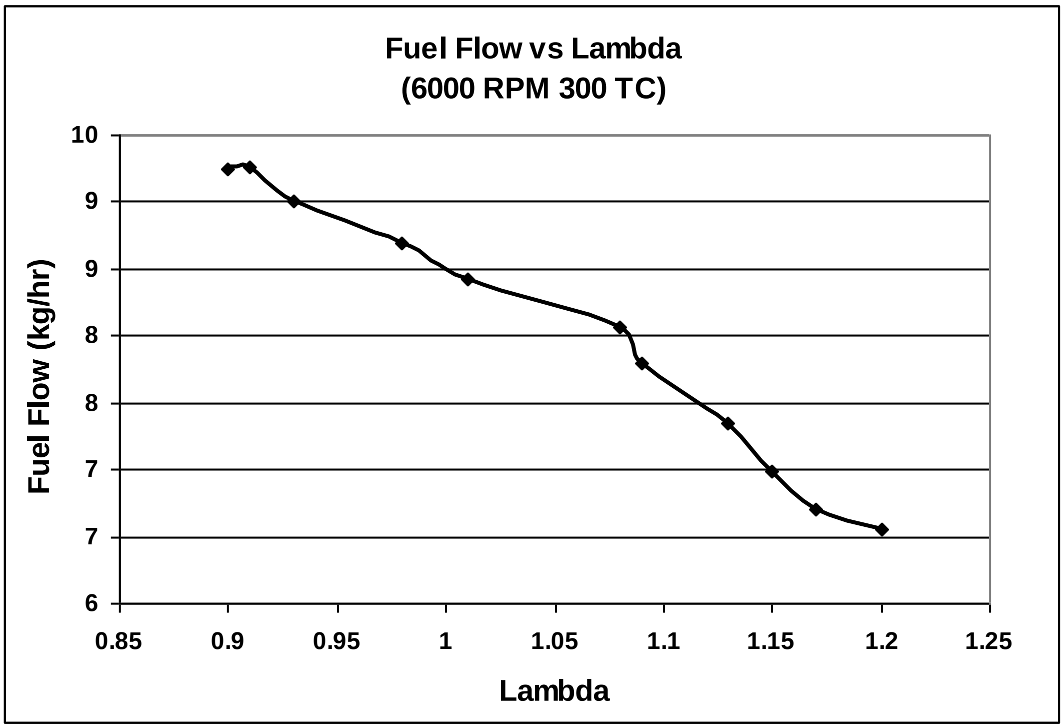

Figure 32: *Fuel flow vs. lambda for 6000 rpm and 30% throttle*

Figures 29, 30, and 31 were presented to help explain some of the fundamental relationships that occur during homogeneous operation and are worth mentioning. While these graphs do convey important concepts, they use lambda as a common relation which is not considered a performance or efficiency parameter and therefore is not accurate when comparing different engine platforms.

If instead, the BSFC and power output vs. lambda graphs are combined, eliminating lambda and producing a graph of power vs. BSFC, we can clearly see the relationship between two engine performance criteria. This graph is shown in Figure 33, which shows the power vs. BSFC data at 6000 rpm and 30% throttle during the fuel sweep. This graph effectively ties all of the important characteristics of homogeneous combustion into one plot and shows that calibrating the engine for peak power does not result in the best BSFC, nor does calibrating the engine at the lean limit of combustion. Instead, it shows the tradeoff in power output and fuel energy conversion efficiency. Keep in mind though that other considerations must be given to emissions and run quality and that calibration for BSFC is not the only focus in engine calibration work.

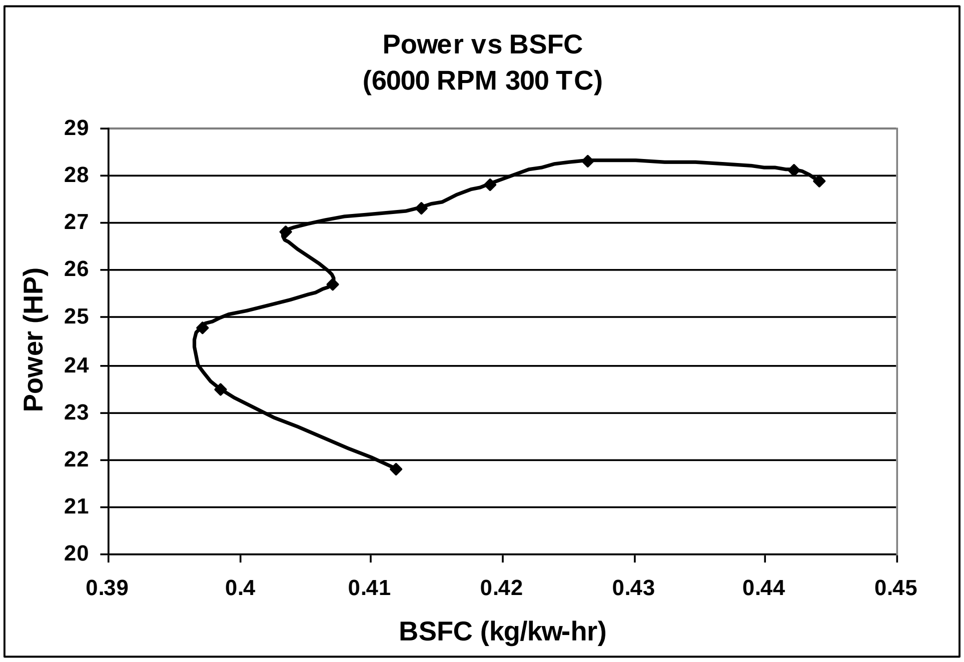

Figure 33: *Power vs. BSFC for 6000 rpm and 30% throttle.*

Looking at Figure 33 more closely, we see that peak power for this calibration point occurs at higher BSFC values due to fuel rich operating conditions and higher flow rates. As the BSFC improves, the power output of the engine decreases. This happens because the fuel flow rate is decreasing with the leaner air-fuel ratios as shown in Figure 32, and even though the power output of the engine is decreasing too Figure 31, it is not decreasing as fast as fuel flow, thus BSFC continues to improve. At the 25 horsepower and 0.398 BSFC point, we see that BSFC has reached a minimum and begins to increase. If we look back at Figures 31 and 32, we see that as the engine is operated at these leaner air-fuel ratios the power output begins to decrease faster than the fuel flow, resulting in a decrease in BSFC.

It is the point of minimum BSFC achieved in the above graph that is of interest in this research. This point is taken to be the most efficient point of operation for the calibration point and the torque produced at this point becomes the target torque that stratified combustion must attain. The BSFC measured during stratified testing is then compared against the one attained during homogeneous testing.

The torques measured at the minimum BSFC points during homogeneous testing are listed in Table 9. These are the target torques that must be achieved during stratified operation in order for stratified combustion to be considered an option for the calibration point. Note that at 4500 and 5000 rpm, the torque output is less than at other engine speeds. This is due to the mistuning characteristics of the exhaust pipe, which in addition to being in a throttled state, delivers irregular plugging pulses to the cylinder causing cyclic variations in residual exhaust gases (EGR). This in turn causes cyclic variations in heat release during combustion and subsequently run quality and power output are negatively affected.

TQ (ft-lb)	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	13.1	11.8	12.0	11.1	13.0	14.1	13.8	13.6
200	17.3	16.3	16.1	15.2	14.7	17.0	19.0	18.2
300	17.8	20.3	19.9	19.8	18.5	19.5	23.6	22.6

Table 9: *Torque achieved at point of minimum BSFC during Homogeneous testing.*

Table 10 shows the minimum BSFC values attained for each calibration point while producing the torque values in table 9. Note that as the throttle is opened and as engine speed increases, BSFC improves. This is due to the reduction in throttling losses resulting in increased torque and power output of the motor for the same engine speed.

BSFC (kg/kW-hr)	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	0.54	0.54	0.43	0.45	0.46	0.43	0.42	0.42
200	0.49	0.48	0.43	0.38	0.44	0.41	0.39	0.40
300	0.48	0.47	0.46	0.39	0.42	0.44	0.40	0.43

Table 10: *Minimum BSFC achieved during homogeneous testing.*

Also, as engine speed increases, the scavenging efficiency improves and the engine is able to generate more torque and power due to reduced EGR, which improves the amount of heat released during combustion. However, as stated earlier, at higher loads and engine speeds the engine is operated fuel rich for maximum power and reduced piston temperatures. As a result, the BSFC at the high load points increases and therefore the point of minimum BSFC for all engine operating conditions occurs somewhere in the higher speed, part load areas. This is due to higher power levels attained at high speed and the ability to run leaner air-fuel ratios due to the throttled condition of the engine, which reduces the heat released during combustion and the concern of piston cooling with excess fuel.

6.2 STRATIFIED

The procedure for stratified testing was performed as outlined in section 5.1.2. For each calibration point the engine was run from the minimum injection angle that could produce torque to the maximum injection angle that still allowed stratified combustion. At each injection angle, the ignition delay was varied from a minimum to a maximum, and then an average delay was tested. BSFC values were recorded for every calibration setting. The goal was to find the point of minimum BSFC for each calibration point, along with finding the ignition delay window and injection timing window that provided stable engine operation.

Figure 34 below is an example plot showing the BSFC curves obtained during testing. The graph is for the 3000 rpm 17 lb-ft target torque (200 TC) calibration point. Plotted is BSFC vs. ignition timing for each injection angle tested at the calibration point, with 50 degrees BTDC being the minimum injection angle capable of producing target torque and 80 degrees BTDC being the maximum injection angle that allowed proper placement of the ignition event, as will be discussed later. The left most point on each series is the maximum ignition delay (late ignition timing), the right most point is the shortest ignition delay (early ignition timing) and the middle point in each series is the average ignition delay.

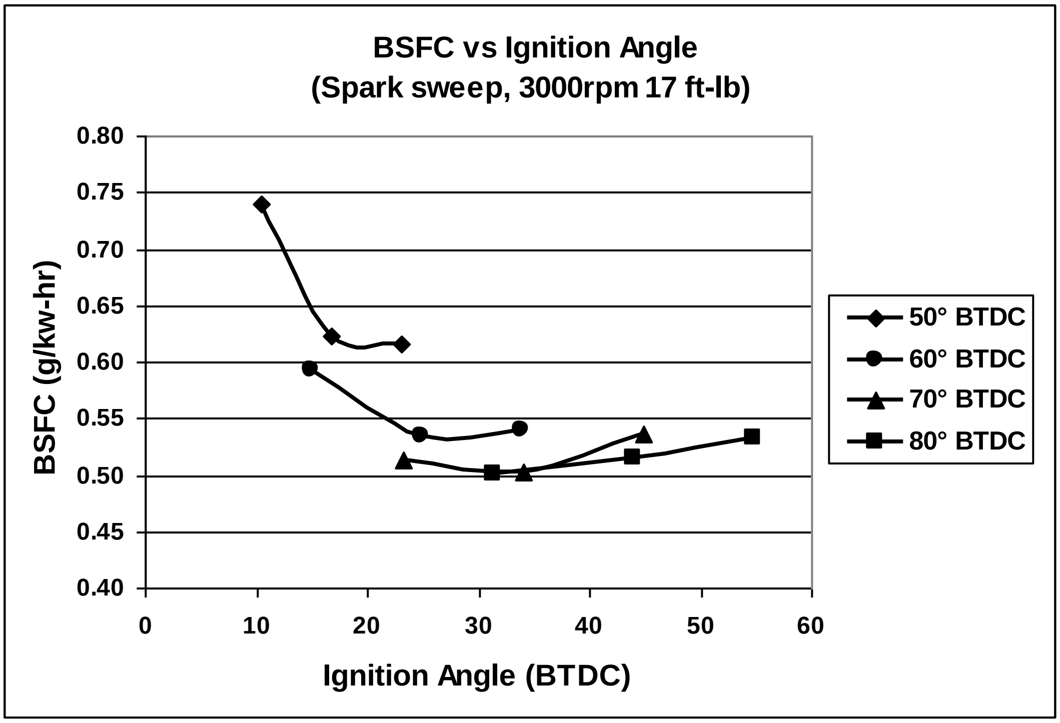

Figure 34: *BSFC vs. Ignition timing for stratified operation. Calibration point is 3000 rpm and 200 TC.*

Note that there is a local minimum BSFC value for each injection angle, but as the injection angle sweep progresses, a minimum BSFC value appears for the entire calibration point. This minimum BSFC value is the purpose of conducting injection angle and ignition delay sweeps, to ensure that the best BSFC for the calibration point is found. It is important to note that although injection angle does affect stratified performance, the ignition delay has a larger effect on engine performance. This is shown in Figure 35 where torque output is plotted against ignition timing for each of the injection angles at 3000 rpm and 200TC.

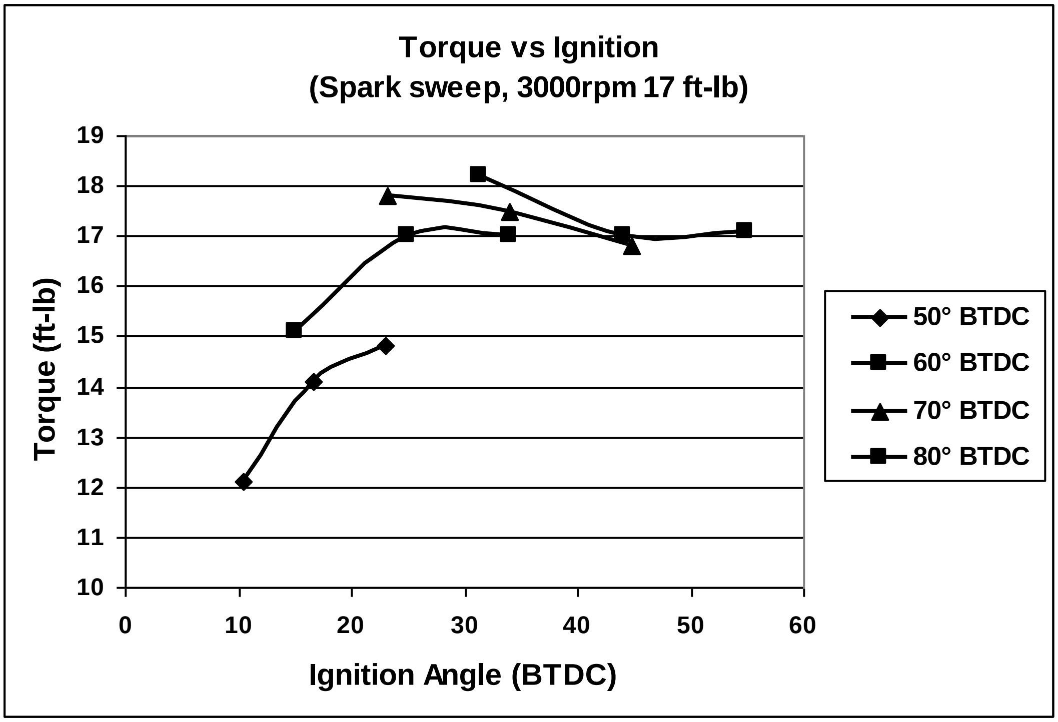

Figure 35: *Torque output vs. Ignition angle for 3000 rpm and 200TC.*

Note that torque increases with both increasing injection angle and ignition angle up to a certain point. The ignition angle during stratified operation is dictated by the ignition delay, which is the time delay, in milliseconds, between the start of injection and the start of the spark event. If we look at the 50 degree injection angle series, we see that torque peaks at about 24 degrees BTDC of ignition timing, which is the minimum ignition delay possible. If the delay is decreased any further, the spark event is occurring before the fuel spray has reached the spark plug and combustion can not start. This short injection angle also limits the amount of time available for air entrainment into the fuel spray, especially during high load stratified conditions where the injected fuel quantity is high. Thus limiting the amount of fuel burned during the cycle and the torque produced. The late ignition timing also delays the start of combustion until later in the cycle, which in turn moves the peak cylinder pressure past the optimum location for maximum torque output. As injection angle is advanced, the same ignition delay values deliver a more advanced spark, starting the combustion process earlier in the cycle and moving the peak cylinder pressure location towards the ideal location for maximum torque. In addition, the advanced ignition timing caused by the advanced injection timing, improves high load stratified operation by allowing a longer period of time for fuel vaporization before the combustion front reaches the last of the injected fuel. Even though the ignition timing is somewhat over-advanced at the peak torque values, Figure 35 shows that for stratified operation, obtaining a complete combustion burn by starting the combustion event sooner and allowing a longer period of time for fuel vaporization is more important than properly phasing the combustion event. During testing it was seen that once the injection angle was advanced above 80 degrees for this calibration point, the minimum ignition delay results in an over advanced spark timing leading to improperly phased combustion and the torque output of the motor decreases. This trend can start to be seen at the 45 and 55 degree ignition timing values in Figure 35.

The concept of ignition delay as a calibration parameter is odd, as it is a time unit. Most calibration parameters are expressed in terms of crank angle degrees before or after top dead center or as a volume of fuel to inject or a pulse width for a fuel injector. Even though ignition delay is an odd parameter for homogeneous calibration, it is useful for stratified calibration. This is because it can be considered as the time available between the start of the injection and spark events that the fuel has to vaporize and mix with the air. If ignition delay is viewed as the mixing time for the fuel, it becomes a useful parameter for stratified combustion. This mixing time concept helps show trends in the time needed for initial fuel vaporization to occur for different calibration points. One such trend is shown in Table 11, which shows the minimum ignition delay values obtained at the 100 TC target torque setting for all of the tested injection angles and engine speeds.

Minimum Ignition Delay 100 TC (msec)							
	Injection Angle						
RPM	***50***	***60***	***70***	***80***	***90***	***100***	***110***
3000	1.3	1.3	1.35	1.3			
3500	1.4	1.3	1.2	1.2			
4000		1.5	1.4	1.4			
4500			1.4	1.1	1.2		
5000			1.2	1.3	1.2	1.2	
5500				1.3	1.1	1.1	
6000				1.25	1.1	1.1	1.1
6500					1.1	0.9	0.9

Table 11: *Minimum Ignition Delay at 100TC target torque vs. injection angle and engine speed.*

Note that for each injection angle, the minimum ignition delay remains fairly constant with engine speed. But as the injection angle is advanced, the ignition delay decreases. This is because the time required for the fuel spray to reach the spark plug is the same regardless of engine speed. However, the time for the fuel spray to reach the spark plug is somewhat dependent on injection angle, as the timing of the injection event determines what cylinder pressure the fuel is injected into and affects the penetration velocity of the fuel and thus the time required for the fuel spray to reach the spark plug. In this case, the more advanced injection angles allow the fuel to be injected into a lower pressure, which increases the penetration velocity and decreases the time for the fuel spray to reach the spark plug.

Another interesting trend is how the minimum ignition delay changes with load or injected fuel quantity. If we look at the minimum ignition delay values for the other two target torque values at 200 and 300 TC (Tables 12&13 respectively), we see that the minimum ignition delay increases with the higher injection quantities, which are required to achieve the higher target torque values.

Minimum Ignition Delay 200 TC (msec)							
200 TC	Injection Angle						
RPM	*50*	*60*	*70*	*80*	*90*	*100*	*110*
3000	1.5	1.45	1.4	1.4			
3500	1.3	1.5	1.4	1.3			
4000		1.3	1.3	1.4			
4500			1.4	1.3	1.35	1.3	
5000			1.3	1.2	1.2	1.2	
5500				1.3	1.2	1.2	
6000					1.3	1.3	1.3
6500					1.3	1.2	1.15

Table 12: *Minimum Ignition Delay at 200 TC target torque.*

Minimum Ignition Delay 300 TC (msec)							
	Injection Angle						
RPM	*50*	*60*	*70*	*80*	*90*	*100*	*110*
3000		1.5	1.45	1.5			
3500		1.4	1.4	1.5			
4000		1.4	1.45	1.5	1.7		
4500				1.6	1.5	1.7	
5000				1.4	1.5	1.3	
5500					1.4	1.2	
6000					1.3	1.4	1.4
6500					1.4	1.35	1.4

Table 13: *Minimum Ignition Delay at 300 TC target torque.*

These larger fuel quantities require longer injection durations because the rate of injection does not change. During stratified combustion, the spark plug is igniting the backside of the fuel spray, and with longer injection durations, there is a longer delay for the back side of the fuel spray to reach the spark plug. Therefore the minimum ignition delay must increase so the ignitable portion of the fuel spray is at the spark plug. Note that for both the 200 and 300 TC calibration points, the minimum ignition delays remain relatively constant with engine speed for any given injection angle, but decrease with increasing injection angle, just like the ignition delays at the 100 TC calibration point. Again this is due to the decreased cylinder pressure and higher penetration velocities at these higher injection angles, which reduce the time required for the backside of the fuel spray to reach the spark plug, thus allowing a shorter minimum ignition delay.

The maximum ignition delay showed similar trends to the minimum ignition delay in that the values seem to be independent of engine speed for any given injection angle but dependent on the injection angle. Tables 14, 15, and 16 show the maximum ignition delay for each injection angle and engine speed at the 100, 200, and 300 TC calibration points respectively. If individual values from the 100 TC, low load table are compared to the higher load tables (200 and 300 TC), we see that the maximum ignition delay increases. This trend occurs because the longer ignition delays provide a longer mixing time for the fuel, allowing the fuel-air mixture to become more of a homogeneous mixture than a stratified one. At the higher fuel quantities, the maximum ignition delay increases because the increased fuel mass in the cylinder reduces the effects of over-mixing caused too long of a mixing time, or ignition delay.

Maximum Ignition Delay 100 TC (msec)							
	Injection Angle						
RPM	***50***	***60***	***70***	***80***	***90***	***100***	***110***
3000	2.2	2	2	2			
3500	1.8	1.7	1.6	1.6			
4000		1.7	1.7	1.7			
4500			1.8	1.7	1.6		
5000			1.7	1.6	1.6	1.6	
5500				1.7	1.5	1.55	
6000				1.7	1.5	1.5	1.4
6500					1.6	1.35	1.4

Table 14: *Maximum Ignition Delay values for 100 TC target torque.*

Maximum Ignition Delay 200 TC (msec)							
	Injection Angle						
RPM	***50***	***60***	***70***	***80***	***90***	***100***	***110***
3000	2.2	2.5	2.6	2.7			
3500	1.7	1.8	2	2			
4000		2	1.9	2			
4500			1.9	1.8	1.8	1.7	
5000			1.8	1.7	1.7	1.6	
5500				1.7	1.7	1.6	
6000					1.8	1.8	1.7
6500					1.6	1.7	1.7

Table 15: *Maximum Ignition Delay values for 200 TC target torque.*

Maximum Ignition Delay 300 TC (msec)							
	Injection Angle						
RPM	***50***	***60***	***70***	***80***	***90***	***100***	***110***
3000		2.6	3	3.2			
3500		2.1	2.2	2.4			
4000		1.9	2.2	2.4	2.7		
4500				2.3	2.3	2.6	
5000				2.1	2.1	2.1	
5500					2	1.8	
6000					1.8	1.9	2
6500					1.6	1.9	1.9

Table 16: *Maximum Ignition Delay values for 300 TC target torque.*

During testing, the run quality of the engine did not suffer at the higher target torque values with the long ignition delays, but torque output decreased due to improper phasing of the combustion event. The negligible effect on run quality occurred because the longer ignition delay allows the larger fuel quantity to mix more thoroughly with the air in the cylinder, thus creating a charge that is semi-homogeneous in the combustion chamber, as previously mentioned. However, run quality at the lower target torque values did suffer as the lower fuel quantities and longer delays allowed for over-mixing of the fuel with in-cylinder air, resulting in erratic and incomplete combustion.

Another factor when using ignition delay is that as engine speed increases and the ignition delay value stays constant, a greater number of crank angle degrees will pass. For example, if a one millisecond ignition delay at 3000 rpm results in the crankshaft rotating 18 degrees, the same one millisecond ignition delay at 6000 rpm will result in the crankshaft rotating 36 degrees. Therefore at the higher engine speeds an earlier injection timing or decreased ignition delay is required to offset the faster rotational velocity of the crankshaft so the combustion event can be phased properly. If possible, the ignition delay should be decreased, but as just discussed, there is a minimum value that this can be, and if that value is reached, the injection angle must be advanced to advance the ignition timing. However, there is a limit to the maximum injection angle before the mixture over mixes with the air in the cylinder and becomes a heterogeneous mixture. This limits the speed at which stratified combustion can occur.

It is interesting to note how ignition delay affects the placement of the spark event relative to the crankshaft position. If we look at the same calibration points as in Table 11, but look at the ignition timing instead of ignition delays, we see that the spark timing becomes very advanced as both injection angle and engine speed increase, as shown in Table 17. Note that at the 50 degree BTDC injection angle and 3000 rpm point the ignition timing is 26.6 degrees BTDC. This corresponds to a 1.1 millisecond ignition delay in Table 11. Now look at the 110 degree BTDC injection angle and 6500 rpm point. The ignition timing is 74.9 degrees BTDC, much more advanced than at the 50 degree injection timing and 3000 rpm point.

Spark Timing 100 TC (Degrees BTDC)							
	Injection Angle						
RPM	***50***	***60***	***70***	***80***	***90***	***100***	***110***
3000	26.6	36.6	45.7	56.6			
3500	20.6	32.7	44.8	54.8			
4000		24	36.4	46.4			
4500			32.2	50.3	57.6		
5000			34	41	54	64	
5500				37.1	53.7	63.7	
6000				35	50.4	60.4	70.4
6500					47.1	64.9	74.9

Table 17: *Spark timing in degrees BTDC for 100TC target torque.*

If we look at the corresponding ignition delay value from Table 11 for the 110 degree injection angle, 6500 rpm point, it is 0.9 milliseconds, only 0.4 milliseconds shorter but the spark timing is advanced almost 50 degrees. Therefore the ignition delay needs to be adjusted any time injection angle or engine speed is changed otherwise the timing of the spark event will not be placed appropriately. This is a key tuning tool that needs to be remembered during stratified engine calibration processes. A table that shows the number of degrees of crankshaft rotation for a given engine speed and ignition delay, such as Table 18, should be used when performing stratified calibration work to easily determine the location of the spark timing in terms of degrees BTDC, which is a more useful value. To calculate the placement of the spark event in terms of degrees BTDC, subtract the value in the corresponding box from the injection timing angle.

Crank Angle Delay (Degrees)	**Ignition Delay (msec)**									
RPM	***0.50***	***1.00***	***1.50***	***2.00***	***2.50***	***3.00***	***3.50***	***4.00***	***4.50***	***5.00***
1000	3	6	9	12	15	18	21	24	27	30
1500	5	9	14	18	23	27	32	36	41	45
2000	6	12	18	24	30	36	42	48	54	60
2500	8	15	23	30	38	45	53	60	68	75
3000	9	18	27	36	45	54	63	72	81	90
3500	11	21	32	42	53	63	74	84	95	105
4000	12	24	36	48	60	72	84	96	108	120
4500	14	27	41	54	68	81	95	108	122	135
5000	15	30	45	60	75	90	105	120	135	150
5500	17	33	50	66	83	99	116	132	149	165
6000	18	36	54	72	90	108	126	144	162	180
6500	20	39	59	78	98	117	137	156	176	195
7000	21	42	63	84	105	126	147	168	189	210
7500	23	45	68	90	113	135	158	180	203	225
8000	24	48	72	96	120	144	168	192	216	240

Table 18: *Degrees of crankshaft rotation as a function of ignition delay and engine speed.*

During stratified testing in the first few calibration points, it was discovered that the target torque could not be achieved during stratified combustion with a throttled engine, as is the case during homogeneous combustion. However, if the throttle plates were opened fully, the target torque was easily achieved. This is because the throttled condition decreased the crankcase pressures and reduced scavenging flow intensities, which affected the placement of the fuel spray in the cylinder at the advanced injection angles and also caused a high amount of EGR to be present. Although combustion occurred and the engine made power, not all of the injected fuel was being burned due to the high EGR content limiting the heat released from combustion. Additionally, the reduced scavenging flows could have allowed the fuel to penetrate further into the cylinder, over penetrating the spark plug and reducing run quality. By opening the throttle plates, higher crankcase pressures could be achieved resulting in more intense scavenging flows which decreased the EGR content in the cylinders and changed the placement of the fuel spray in the cylinder. It is not known whether the decrease in EGR content or the possibility of the increased scavenging flows re-locating the fuel spray to a more favorable position for stratified operation was the source of improved run quality.

Therefore, all stratified data points collected during testing were performed with the throttle plates set at wide open throttle. The torque output of the motor was controlled with fuel quantity, injection angle and ignition delay, and values were adjusted accordingly to achieve the target torques for each calibration point. The following is the presentation of the torque and BSFC values achieved during stratified operation.

Table 19 shows the torque achieved for each calibration point during stratified operation. Note that these values are the same as those achieved during homogeneous testing as shown in Table 20.

TQ (ft-lb)	**RPM**							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	13.0	12.7	12.4	11.2	13.0	15.0	14.4	14.1
200	17.5	16.1	16.0	15.4	15.0	17.6	19.0	18.8
300	18.7	20.3	19.0	19.9	18.2	19.5	22.5	22.7

Table 19: *Observed torque during initial stratified calibration.*

% Target TQ	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	99	108	103	101	100	106	104	104
200	101	99	99	101	102	104	100	103
300	105	100	95	101	98	100	95	100

Table 20: *Percent of homogeneous target torque achieved during stratified operation.*

Table 20 shows the percent of the homogeneous target torque achieved during stratified testing. As can be seen, stratified combustion was able to match the torque achieved during homogeneous combustion for all of the points tested, verifying that stratified combustion can be considered as an alternative to homogeneous combustion from a power output standpoint at the cruise points of the engine.

The minimum BSFC values obtained during stratified testing at each calibration point are presented in Table 21. Note that there is no apparent trend in the values to load or engine speed. This might be due to improper testing methods or the fact that injected fuel quantity, fuel flow, and torque output during stratified operation are all directly and linearly related, as will be shown later in section 6.3.

BSFC (kg/kW-hr)	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	0.52	0.48	0.45	0.49	0.49	0.46	0.43	0.47
200	0.50	0.46	0.45	0.42	0.46	0.43	0.44	0.43
300	0.56	0.47	0.49	0.43	0.44	0.43	0.48	0.47

Table 21: *Minimum stratified BSFC achieved while maintaining homogeneous target torque.*

Table 22 compares the BSFC values for stratified data to those obtained during homogeneous testing. The values are the percent change in BSFC that stratified combustion provided over homogeneous combustion, and values are compared on a cell by cell basis. In this table, a negative number represents a better or improved BSFC while a positive number indicates a worse BSFC.

% Diff in BSFC	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	-3.8	-12.1	3.8	7.6	6.1	5.0	3.6	11.6
200	2.1	-4.2	3.6	10.8	4.5	5.1	11.1	7.1
300	14.9	-0.8	5.9	9.9	5.5	-1.7	15.5	7.8

Table 22: *Percent change in BSFC between homogeneous and stratified. Positive is an increase, negative a decrease.*

Note that BSFC improved only in a few places for stratified combustion, with most of the calibration points indicating that homogeneous combustion is a more efficient mode of combustion at these points. This was expected at the higher loads and engine speeds, such as 300 TC and 5000 rpm and faster engine speeds. However, at the lower loads and engine speeds, such as below 200 TC and 4000 rpm, it was expected that stratified combustion would be better, which is indicated somewhat by the negative values at 3000 and 3500 rpm engine speeds.

6.3 HIGH SQUISH STRATIFIED

The high squish stratified testing was performed similarly to the procedure outlined in section 5.1.2; however a slight change was made to decrease the amount of time required to gather data. An injection angle sweep was performed as outlined, but instead of doing ignition delay sweeps, a fuel sweep was done. When the correct fuel quantity to achieve the target torque value was found, the minimum and maximum ignition delays were noted and the fuel sweep continued. This process was done to speed up the testing process and also to make the search for trends in relations between injection angle, fuel quantity, torque, and BSFC easier. For the data being collected, it did not matter in which manner it was collected so long as the engine and equipment were given sufficient time to reach a steady state condition and stabilize.

Table 23 contains the data obtained for stratified operation with modified cylinders, starting with the torque achieved at each of the calibration points

TQ (ft-lb)	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	13.3	12.1	12.3	11.0	12.8	14.0	13.8	13.5
200	16.6	16.1	16.1	14.8	14.9	17.2	18.9	17.2
300	17.9	19.0	20.3	19.3	17.9	18.3	18.9	17.2

Table 23: *Torque achieved during high squish stratified operation.*

Note that peak torque produced remains relatively constant among all engine speeds, which is in contrast to the torque achieved during homogeneous and initial stratified testing. Comparing the torque values in Table 23 to those achieved during homogeneous testing we get the values in Table 24, which shows the percent of homogeneous target torque achieved during high squish stratified operation.

% Target TQ	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	102	103	103	99	98	99	100	99
200	96	99	100	97	101	101	99	95
300	101	94	102	97	97	94	80	76

Table 24: *Percent of homogeneous target torque achieved during high squish stratified operation.*

Note that at the 100 and 200 TC target torque settings the high squish stratified testing resulted in acceptable torque output when compared to the homogeneous testing. However, at the 300 TC target torque setting the torque produced at 5500 is 94% of the target torque and at 6000 and 6500 rpm the torque achieved is only 80% and 76 % of the target torques respectively. Therefore the 6000 and 6500 rpm points are highlighted to indicate that the torque produced was not sufficient to meet the target torque set during homogenous operation, and therefore stratified operation can not be considered as an alternative to homogeneous combustion at these points. For the rest of the data presented, a highlighted box indicates that the target torque was not achieved and the values should be disregarded, but are presented for completeness.

The decrease in torque output at the high load points was not expected, as it was thought that increasing squish velocity would increase the torque output during high load stratified operation due to improved fuel-air mixing. However, this was not the case. If we look at the BSFC data collected, (Table 25), we see that BSFC is not mostly independent of load and engine speed, the same trend that was seen during unmodified stratified testing.

BSFC (kg/kW-hr)	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	0.53	0.50	0.47	0.52	0.50	0.48	0.49	0.51
200	0.52	0.51	0.46	0.51	0.48	0.48	0.46	0.47
300	0.54	0.55	0.49	0.49	0.49	0.51	0.46	0.47

Table 25: *BSFC values for high squish stratified operation.*

This is because during stratified operation, the torque output and subsequently power output, quantity of fuel injected into the cylinder, and fuel flow to the motor are directly and linearly related. Therefore, as load increases, (increased torque and power) so does the fuel quantity and fuel flow, thus allowing BSFC to remain fairly constant.

Table 26 shows the percent difference in BSFC between high squish stratified and homogeneous operation. As before, a positive number indicates an increase in BSFC while a negative number indicates an improvement. This shows that only the 3000 and 3500 rpm and 100 TC torque points are better with high squish stratified operation, which was similar for stratified operation with the unmodified cylinders (Table 22).

% Diff in BSFC	RPM							
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	-2.1	-8.6	8.3	13.8	8.6	9.5	15.1	18.2
200	5.0	5.0	5.7	26.9	8.7	15.2	15.4	14.5
300	12.2	13.7	7.3	20.7	15.3	13.9	13.0	8.9

Table 26: *Percent change in BSFC between high squish stratified and homogeneous.*

However, if we look at the rest of the values in the table, there are values that are 15% to 25 % higher than obtained during homogenous combustion. Indicating that high squish stratified operation is not as efficient as homogeneous calibration. But was there an improvement made to stratified operation? Table 27 compares the stratified BSFC values obtained with the modified cylinders to those values obtained with the unmodified cylinders. Again a negative number indicates an improvement or reduction in BSFC while a positive number indicates a worse BSFC.

% Diff in BSFC	RPM						
TP	***3000***	***3500***	***4000***	***4500***	***5000***	***5500***	***6000***	***6500***
100	1.6	3.2	4.7	6.8	2.7	4.8	12.0	7.5
200	2.9	8.9	2.2	18.0	4.3	10.6	4.8	8.0
300	-3.1	14.4	1.4	12.0	10.4	15.3	-3.0	1.2

Table 27: *Percent change in BSFC between high and low squish stratified operation.*

As can be seen, there was only one point that high squish stratified operation improved BSFC values, and that is at 3000 RPM and 300 TC. At all the other calibration points, increasing squish velocity decreased the performance and efficiency of stratified operation substantially.

6.4 CONCLUSIONS

This research set out to determine three major things: first was to determine whether stratified combustion or homogeneous combustion provided better fuel economy in the cruise point range of engine operation. Second was to determine where the transition from stratified combustion to homogeneous combustion should occur in the engine map. Third was to see how increased squish velocity affected stratified operation. The testing performed yielded good results, both expected and unexpected.

These data clearly show that homogeneous calibration is the superior mode of combustion for the cruise points of a snowmobile engine. The first indicator of this is the fact that stratified combustion could not produce any of the target torque values unless the throttle plates were held at WOT. This increased scavenging performance and reduced the amount of EGR in the cylinder during combustion, which helped ensure that the maximum possible heat release could be attained during combustion. The second benefit of having the throttle plates at WOT was that the more intense scavenging flows affected the placement of the fuel spray at the advanced injection angles, perhaps moving the fuel spray to a more favorable place for stratified operation. The second indicator that stratified combustion was inferior to homogeneous combustion in the cruise points was the increased BSFC values, which indicate lower overall engine efficiencies, as explained in section 5.1 by equation 5.1.3, which related the BSFC inversely to the heating value of the fuel and overall efficiency of the engine.

The transition from stratified to homogenous combustion does not appear in the data collected. There is no definitive evidence that stratified combustion is better than homogeneous combustion in the calibration points tested above. Initially it was thought that this set of calibration points was broad enough that they would include a particular region where the BSFC values for homogeneous and stratified combustion were the same, indicating a transition from one mode of combustion to the other. However, no such trend was found with these data. A suggestion as to the placement of the transition is presented later in section 7.1.

Results for the third goal of this thesis were also obtained, although they were not what were expected. Increasing the squish velocity was shown to hinder the torque output of the engine at 6000 and 6500 rpm at the 300 TC points, which are the high speed high load points during this testing. Increasing the squish velocity was thought to improve stratified operation at these points with improvements to both the peak torque produced during stratified operation and the BSFC values and overall engine efficiency. This however was not the case. The increased squish velocity decreased the peak torque produced and worsened the BSFC values over the ones obtained with lower squish velocity stratified operation.

There are several reasons that have been hypothesized for this trend and they are presented here. However, without detailed in-cylinder combustion data, no definite explanation can be offered as to why this trend appears. The first possible reason for the decrease in the maximum injection quantity for high squish stratified operation is the effect the squish flows have on the fuel spray during the combustion event. During high speed stratified operation, early injection angles and early ignition timings are used to start the combustion process early in the cycle. This is because of the time required to initiate a flame front during stratified operation (0-10% mass burn duration) as it is a finite process and contributes to the timing and amount of heat released during the combustion event. Stratified combustion usually has a longer 0-10% burn duration than homogeneous combustion, which justifies the overly advanced ignition timing. But despite the advanced ignition timing, the main part of combustion (10-90% burn duration) occurs while the piston is at or near top dead center, which is when the effects of squish contribute to the combustion process. There are two theories that arise out of this.

The first theory is that the higher squish velocities at the higher engine speeds are fast enough to extinguish combustion. At the point that squish velocity affects the combustion event, there may not be a fully developed and stable flame at the high load points with higher fuel quantities. This is because of the longer ignition delays required for the ignitable portion of the fuel spray to reach the spark plug, which delays the onset of the main combustion event until later in the cycle, and thus the flame front at the start of the squish process might not be stable enough for a sudden change in cylinder motion. Thus the flame would be susceptible to being blown out by the squish velocities, stopping the combustion process all together. However, this would not happen at the lower load points and fuel quantities as the shorter ignition delays start the combustion process sooner, allowing a longer period of time for the flame front to develop from the initial burn stage into the main burn stage with a more robust flame front, decreasing the susceptibility of the flame to squish velocity and the possibility of extinguishing the flame [19].

The second theory for the decrease in the maximum injection quantity for high squish stratified operation at the higher engine speeds and loads is that the higher squish velocity is pushing the injected fuel into the top, main part of the chamber, preventing it from penetrating further into the combustion chamber and hindering air entrainment into the portion of the fuel spray at the top of the chamber. This would create a locally rich zone in the combustion chamber incapable of supporting combustion and resulting in an extinguished flame. This theory is supported by the fact that any increases in fuel quantity result in further reductions in run quality and torque output, indicating an even richer condition in the local area. If the increased squish velocities were promoting the entrainment of air into the fuel mixture, the addition of fuel should have increased both the maximum amount of fuel that could be injected and the maximum producible torque during stratified operation. However, this is not the trend that is observed. A possible solution to this problem, if it were proven true, would be to target the squish motion into a better location. By redesigning the squish band on the cylinder head and reshaping the piston to redirect the squish motion from the squish band up into the main part of the chamber along the chamber walls, air entrainment into the backside of the fuel spray would be possible. This might prove to be one possible solution [19, 20].

Again, without in-cylinder combustion data or emissions equipment, an explanation for the trends found in this data can only be theorized. The results do show that increasing the squish velocity in a spray guided GDI system with an outwardly opening nozzle and hollow cone of fuel decreases the performance of stratified combustion. Several theories have been presented and can invoke thoughtful discussions, but a definitive explanation cannot be provided at this time.

7.0 RECOMMENDATIONS AND FUTURE WORK

This section presents a brief recommendation of where future research should be directed. The development of engine technology is complex and time consuming. There are many possible research areas but those presented here should be investigated as the next steps to take in GDI two stroke engine research at the University of Idaho.

7.1 STRATIFIED TO HOMOGENEOUS TRANSITION

Based on the results of the testing performed, it is recommended that stratified operation be limited to engine speeds of 3000 rpm or less and throttle settings less than 10%. Figure 36 below is a suggestion of where the transition from stratified to homogeneous operation should occur.

Throttle Position	Engine Speed										
	500	1000	1500	2000	3000	4000	5000	5500	6000	7000	8000
10											
50			Stratified								
80											
100			Transition								
150											
200											
250											
300											
301						Homogeneous					
400											
500											
600											
700											
800											
1000											

Figure 36: *Proposed location for stratified to homogeneous transition.*

Note though that this serves as a guideline and such things as engine run quality and throttle response in the transition region will ultimately dictate where this transition should occur. This figure should be considered as a guideline for initial calibration strategies for future engine development.

7.2 STRATIFIED COMBUSTION

Future work for stratified combustion should be focused on determining the correct spark plug length to properly locate the spark plug in the fuel spray. This was not done during this research, but may turn out to improve the stratified combustion significantly as stratified combustion is sensitive to this parameter. Another area of research would be the investigation of targeting the squish motion to see where the squish should be directed to improve stratified operation. This could be accomplished through incrementally increasing the radius between the squish band and the main chamber area to promote the squish motion to follow the walls of the cylinder head as opposed to the piston surface. Another way to accomplish the same effect is to implement a small dome to the top of the piston that is the same shape of the bottom portion of the main chamber, which would help direct the squish motion up along the walls of the main chamber and into the backside of the fuel spray. Additionally, the performance of stratified combustion with E-85 as a fuel should be investigated and a comparison of any differences be made to the data presented here.

7.3 ENGINE MODELING

Future research should also be directed at establishing a computer model of the engine capable of in-cylinder flow visualizations at multiple engine speeds. Such a model should include visualization of scavenging flows through the transfer ports, fuel spray penetration into the chamber and squish motion. This model can show each of these independently for ease of modeling, but a model capable of showing all three flows simultaneously would prove to be a great asset. Future combustion chamber shapes and possibly piston designs could be based from insights gained from visualizations of the in-cylinder flow fields. The only downfall to accomplishing a model capable of this is the verification process with actual data and the acquirement of said data.

7.4 LIGHT LOAD HOMOGENEOUS COMBUSTION

Future research should be directed towards the improvement of light load, homogenous combustion. As discussed in section 6.1 and shown in figures 30, 31 and 32, the efficiency of the engine increases at leaner air fuel ratios until the lean limit of combustion is reached and high misfire rates lead to reduced power output, increased BSFC and reduced overall efficiency. Reducing the misfire rate at the leaner air-fuel ratios by will improve power output and further improve efficiency, BSFC, and ultimately fuel economy. Additionally, the mistimed pressure waves caused by the tuned exhaust pipe when the engine is operated at off-design speeds were also verified to decrease power and torque output of the engine as shown in table 9 by the decrease in torque output at 4500 and 5000 rpm when compared to the results at 4000 and 5500 rpm respectively.

One method to improve homogeneous combustion at the leaner air-fuel ratios is to design a cylinder head that utilizes two spark plugs per cylinder. The reason for increased misfire at lean air-fuel ratios is due to an increase in the required ignition energy to initiate combustion of the fuel-air mixture. Utilizing two spark plugs per cylinder will increase the energy input into the fuel-air mixture as well as provide a second location for combustion to initiate, which should lead to reduced misfire rate and improve power output.

One method to reduce the effect of the mistimed exhaust pulses is to explore the use of exhaust valves between the exhaust ports and tuned pipe. The use of exhaust valves, which are essentially a throttle plate in the exhaust, will help to dampen out the mistimed pressure waves caused by the exhaust pipe when the engine is operated at off design speeds. Dampening out these waves will reduce the cycle-to-cycle variations in EGR content in the cylinder, which in turn will reduce the cycle-to-cycle variations in heat release from combustion and ultimately improve run quality and power output. Research should also be done to investigate the possibility of running an engine without throttle plates, relying on the exhaust valves to provide a steady amount of EGR for any given engine speed and thus limiting the heat release from combustion, and ultimately the torque output of the engine.

However, before any further research is conducted towards the development of GDI two-stroke engine technology, higher quality engine testing equipment is needed. A dynamometer that is capable of holding engine speed to within 15 rpm of a desired engine speed regardless of changes in torque output of the motor would add to the accuracy of the data. A fuel measuring system capable of steady state flow readings that deviate less than 2% from the true flow reading is also necessary. Additionally an emissions analyzer capable of continuously reading emissions from an un-calibrated engine for up to 10 minutes at a time would greatly shorten the amount of time required for emissions tuning and engine mapping. In-cylinder pressure data with the ability to analyze combustion data immediately after sampling would also reduce the time needed to calibrate stratified combustion as well as provide insights to subtle changes in engine calibration parameters such as ignition delay and injection angle during stratified combustion.

BIBLIOGRAPHY:

1. Society of Automotive Engineers, Inc., The SAE Clean Snowmobile Challenge Rules 2006, (www.sae.org/competitions/snow), April 3, 2007.

2. 69 Fed.Reg. 65,348 (Nov. 10, 2004).

3. Strauss, S., Zeng, Y., Montgomery, D., "Optimization of the E-TEC™ Combustion System for Direct-Injection Two-Stroke Engines Toward 3-Star Emissions," SAE 2003-32-0007/20034307, 2003.

4. Beach, N., et al. "University of Idaho's Clean Snowmobile Design Using a Direct-Injection Two-Stroke with Exhaust After Treatment," Not Published.

5. Bradbury, N., Schiermeier R., Harris, T., "University of Idaho's Clean Snowmobile Design Using a Direct-Injection Two-Stoke," SAE 2005-01-3680, 2005.

6. Bradbury, N., et al. "University of Idaho's Clean Snowmobile Design Using a Direct Injection Two-Stroke," SAE 2006-32-0050, 2006.

7. Morikawa, K., Takimoto H., Ogi T., "A Study of Direct Fuel Injection Two-Stroke Engine for High Specific Power Output and High Engine Speed," SAE 1999-01-3288 /JSAE 9938043, 1999.

8. Heywood J.B., *Internal Combustion Engine Fundamentals.* McGraw Hill, Inc. 1988.

9. Bradbury, N., "Retrofitting Direct-Injection and a Turbocharger to a Two-Stroke Engine for Snowmobile Applications," M.S. Thesis, University of Idaho, 2006.

10. Moriyoshi, Y., Mori, K., "Mixture Formation Analysis of a Schnurle-Type Two-Stroke Gasoline DI Engine," SAE 2001-01-1091, 2001.

11. Zhao, F., Harrington, D., Lai, M., *Automotive Gasoline Direct-Injection Engines*. SAE Inc. Warrendale, Pa. 2002.

12. Findlay, A., et al. "University of Idaho's Clean Snowmobile Design Using a Direct-Injection Two-Stroke Engine," 2007, not yet published.

13. Strauss, S., Zeng, Y., "The Effect of Fuel Spray Momentum on Performance and Emissions of Direct-Injected Two-Stroke Engines," SAE 2004-32-0013/20044300, 2004.

14. Blair G.P. *Design and Simulation of Two-Stroke Engines*. Society of Automotive Engineers, Inc. Warrendale, Pa, 1996.

15. Szekely, G., Alkidas, A., "Combustion Characteristics of a Spray-Guided Direct-Injection Stratified-Charge Engine with a High-Squish Piston," SAE 2005-01-1937.

16. Schanzlin, K., et al. "Characterization of Mixture Formation in a Direct Injected Spark Ignition Engine," SAE 2001-01-1909, 2001.

17. Stone, R., *Introduction to Internal Combustion Engines*. Antony Rowe Ltd. Chippenham, Wiltshire, 1997.

18. Land and Sea, Inc., DynoMite™ Snowmobile Dyno #075-200-1K Specs, (http://www.land-and-sea.com/snowmobile-dyno/DYNOmite%20Snow%20075-200-1K%20Quick-Specs.pdf), April 16, 2007.

19. Nehmer, Dan. BRP Inc., Personal correspondence, 4/20/07.

20. Montgomery, Dave. Caterpillar Inc., Personal correspondence, 4/24/07.

APPENDIX A – Squish Calculator Code (MatLab)

```
%Engine Geometry - (mm)

n = 1;                          %Number of Cylinders
d_bo = 77.24;                      %Bore Diameter
d_b = 49.53;                       %Diameter of Inner Chamber
L_st = 64;                       %Stroke
L_cr = 124;                       %Rod Length
Ex = 102;                        %Exhaust Port Opening (deg)ATDC
In = 120;                        %Intake Port Opening (deg) ATDC
L_ct = 0.5*L_st;                   %Throw of the crank
A_p = pi*(d_bo/10/2)^2;               %Area of the Piston
V_chamber = 40.62;                   %Chamber Volume (cm^3)

A_squish = pi*((d_bo/20)^2-(d_b/20)^2)  %Squish Area (cm^2)
x_p = input('Enter the Squish Clearance in inches')             %Squish clearance (cm)
x_s = x_p*2.54;
Squish_Clearance = x_s;

V_squish = Squish_Clearance*A_squish;
V_squish                         %Squish Volume

V_inner_chamber = V_chamber-4.550;
V_cv = V_inner_chamber+V_squish;

=================================================
%Calculations

L_ts = L_cr + L_ct*(1-cosd(Ex)) - (L_cr^2 - (L_ct*sind(Ex))^2)^0.5;     %Trapped Stroke
Trapped_Stroke = L_ts


V_sv = (n*(pi/4)*d_bo^2*L_st)/1000;       %Total Swept Volume (cm^3)
Swept_Volume = V_sv


V_ts = (n*(pi/4)*d_bo^2*L_ts)/1000;       %Total Trapped Volume (cm^3)
Trapped_Volume = V_ts


CR_g = (V_sv*(1/n)+V_cv)/V_cv;          %Geometric Compression Ratio
Geometric_Compression_Ratio = CR_g


CR_t = (V_ts*(1/n)+V_cv)/V_cv;          %Trapped Compression Ratio
Trapped_Compression_Ratio = CR_t


Theta = (0:359);                   %Crank Angle
Y=(L_cr/1000)*ones(1,360);
```

APPENDIX A – Squish Calculator Code (MatLab) cont'd

```
H_theta = (L_cr/1000 + (L_ct/1000)*(1-cosd(Theta))-(((Y.^2)-
(((L_ct/1000)^2).*sind(Theta)).^2).^.5));       %Location of piston from top dead center (m) (0 deg
is TDC)


V_cyl_theta = (H_theta*100).*A_p + V_cv;     %Volume of cylinder (cm^3)

====================================================

%Calculations for Squish Velocity

RPM =[3000;4000;5000;6000]               %Engine Speed
rpm_1 = RPM(1,1)/60;           %deg/sec
rpm_2 = RPM(2,1)/60;
rpm_3 = RPM(3,1)/60;
rpm_4 = RPM(4,1)/60;
d_theta = 1;



C_sq = A_squish/((pi/4)*(d_bo/10)^2);  %Squish area ratio
Squish_Area_Ratio = C_sq

V_s_theta = (V_squish + A_squish*H_theta*100);     %Volume of walls of a hollow cylinder
extending from the squish band to the top of the piston
V_b_theta = (V_cyl_theta - V_s_theta);             %Volume of the hollow in the cylinder
described above


A_sq = pi*(d_b/10)*(x_s+(100*H_theta(1,349)+100*H_theta(1,348))/2);     %Squish area
perpendicular to cylinder bore (cm^2)

c_sq1 = (V_cyl_theta(1,349)/A_sq)*[(V_s_theta(1,348)/V_cyl_theta(1,348)) -
(V_s_theta(1,349)/V_cyl_theta(1,349))]*360*rpm_1/d_theta;

Squish_1 = c_sq1/100;

====================================================

%All Possible Squish Velocities

Toa = (1:360);

H_Toa = L_cr/1000 + (L_ct/1000)*(1-cosd(Toa))-(((Y.^2)-(((L_ct/1000)^2).*sind(Toa)).^2).^.5);
%Height of piston in T angle (m)

V_s_toa = V_squish + A_squish*H_Toa*100;
V_cyl_toa = ((100*H_Toa)*A_p + V_cv);

A_s_Theta_Toa = pi*(d_b/10)*(x_s+(100*H_theta + 100*H_Toa)/2);
```

APPENDIX A – Squish Calculator Code (MatLab) cont'd

```
j=(1:1:360);     %j goes with Toa

===================================================
%First RPM point
c_sq_Theta_Toa_1(1,j) = (V_cyl_toa./A_s_Theta_Toa).*[(V_s_theta(1,j)./V_cyl_theta(1,j))-
(V_s_toa(1,j)./V_cyl_toa(1,j))].*360*rpm_1/d_theta;

Squish_Velocity_1 = c_sq_Theta_Toa_1/100;
Max_Squish_Velocity_1 = max(Squish_Velocity_1)

%Second RPM Point
c_sq_Theta_Toa_2(1,j) = (V_cyl_toa./A_s_Theta_Toa).*[(V_s_theta(1,j)./V_cyl_theta(1,j))-
(V_s_toa(1,j)./V_cyl_toa(1,j))].*360*rpm_2/d_theta;

Squish_Velocity_2 = c_sq_Theta_Toa_2/100;
Max_Squish_Velocity_2 = max(Squish_Velocity_2)

%Third RPM Point
c_sq_Theta_Toa_3(1,j) = (V_cyl_toa./A_s_Theta_Toa).*[(V_s_theta(1,j)./V_cyl_theta(1,j))-
(V_s_toa(1,j)./V_cyl_toa(1,j))].*360*rpm_3/d_theta;

Squish_Velocity_3 = c_sq_Theta_Toa_3/100;
Max_Squish_Velocity_3 = max(Squish_Velocity_3)

%Fourth RPM point
c_sq_Theta_Toa_4(1,j) = (V_cyl_toa./A_s_Theta_Toa).*[(V_s_theta(1,j)./V_cyl_theta(1,j))-
(V_s_toa(1,j)./V_cyl_toa(1,j))].*360*rpm_4/d_theta;

Squish_Velocity_4 = c_sq_Theta_Toa_4/100;
Max_Squish_Velocity_4 = max(Squish_Velocity_4)

=================================================
plot(Toa,Squish_Velocity_1,Toa,Squish_Velocity_2,Toa,Squish_Velocity_3,Toa,Squish_Velocity_
4)
xlabel('Crank Angle')
ylabel('Squish Velocity')
legend('3000','4000','5000','6000')
title('Squish Velocity vs. Crank Angle for Different RPMs')
```